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Buying Coal

By R. T. STROHM

WHEN business was largely a matter of bluff,
And cunning and fraud and deception held sway,
The purchase of fuel was simple enough,
And bore less resemblance to work than to play.
We wired the dealer, or maybe we wrote,
"Deliver two carloads by August the first,"
And added, in hope of annexing his goat,
"That last lot you sent us was surely the worst."

The stuff that he shipped us was doubtlessly coal,
In spite of its heavy proportion of slate.
We managed to use it, but out of the whole,
One-third was unburnable, simply dead weight.
The grates clogged with clinkers, the firemen raved,
And firing became a Tartarean dream,
Yet nobody cared how those fellows behaved,
So long as they kept up the pressure of steam.

But guesswork and heedlessness now are disgraced,
While care and efficiency reign in their turn,
And owners desirous of cutting out waste
Keep mighty close watch on the stuff that they burn.
They test every shipment of coal that they get—
They're all from Missouri—they've got to be shown;
And no tricky dealer is able, you bet,
To stick them with coal that is half clay and bone.

The specifications they've drawn clearly show
The heat that the fuel must yield under test,
As well as the limits to which it may go
In moisture and gases and ash and the rest.
And so, if a dealer should stoop, on the sly,
To sprinkle his shipment to add to its heft,
Or hope with a low grade of stuff to get by,
The chances are fifty to one he'd get left.

A New Departure in Ammonia Compression

By Charles H. Bromley

SYNOPSIS—A high-speed feather-valve air compressor with compressor details redesigned for ammonia is connected into the suction line of a 400-ton compressor to provide the latter with suction gas at 40 to 60 lb. pressure instead of gas at 5 lb., as formerly. The cost per ton of the increased capacity is thus made very low. The system has ammonia and steam details that are of unusual interest.

One naturally looks for pioneer work in a system that furnishes refrigeration, light and power to what is the most intensive, if not the largest, market place on earth.

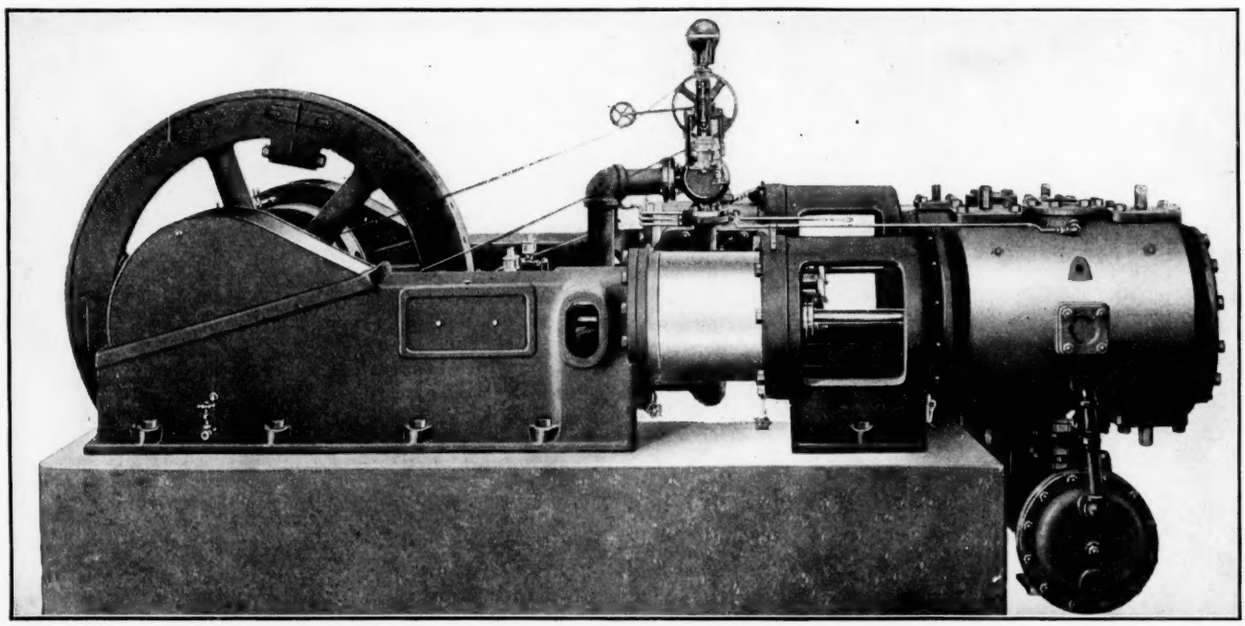


FIG. 1. HIGH-SPEED DUPLEX BOOSTER FOR AMMONIA COMPRESSION

This summer will see the street and warehouse systems of the Quincy Market Cold Storage and Warehouse Co.,[1] Boston, so heavily loaded that greater capacity must be provided. It is the method of obtaining this additional capacity that is novel and likely portentous.

In the company's Richmond Street station there is a 400-ton ammonia compressor cooling brine for a warehouse system. The brine is maintained at between —10 and —12 deg. F., which requires a suction-gas pressure of between 7 and 8 lb. gage, Goodenough's tables (actual operating pressure about 5 lb. gage). The discharge, or condenser, pressure varies between 120 and 185 lb., depending upon the temperature of the cooling water. This compressor was rebuilt some years ago and its speed increased to a safe maximum of 75 r.p.m. The station is so crowded that it does not permit installing another large compressor. There is plenty of water-pumping capacity, and the condenser room contains hundreds of tons' capacity in submerged condensers now made available since the installation some time ago of a jet-type ammonia condenser.[2]

Alongside of the 400-ton compressor is a 250-ton absorption system. The conditions, therefore, are well suited to the new two-stage compression method, which is obtained as follows:

A high-speed, 250 r.p.m., 13x16-in. steam end, 20x16-in. ammonia end, feather-valve air compressor, Fig. 1, with its details adapted to ammonia, is connected in the suction line of the 400-ton machine, as shown in Fig. 2. This compressor, or booster, will take the gas from the coolers at 5 lb. gage and increase it to a maximum of 60 lb. gage for compression in the large machine to the condenser pressure. In the four cold months with condenser pressure under 100 lb., the booster will discharge to condensers directly, cutting out the large machine. Fig. 3 shows a part section of the ammonia cylinder of the booster.

Consider what handling the low-pressure vapor at high speed means in weight of ammonia circulated by both compressors as compared to the weight handled in the same time under old conditions. The difference will be proportional to the difference in absolute pressures of the suction gas; the pressures used are therefore absolute. The vapor returns from the coolers at 20 lb. pressure, —15.9 deg. F., actually —8 to —10 deg. (slight superheat). The yearly mean condenser pressure is, say 150 lb. The effective displacement of the large machine, which has two single-acting ammonia cylinders, is 12 cu.ft. per

[1]For description of the system and its equipment by the author, see "Power," Dec. 8, 15, 22, 29, 1915, and Jan. 2, 1916.

[2]For description of this condenser, see "Power," Mar. 21, 1916, p. 390.

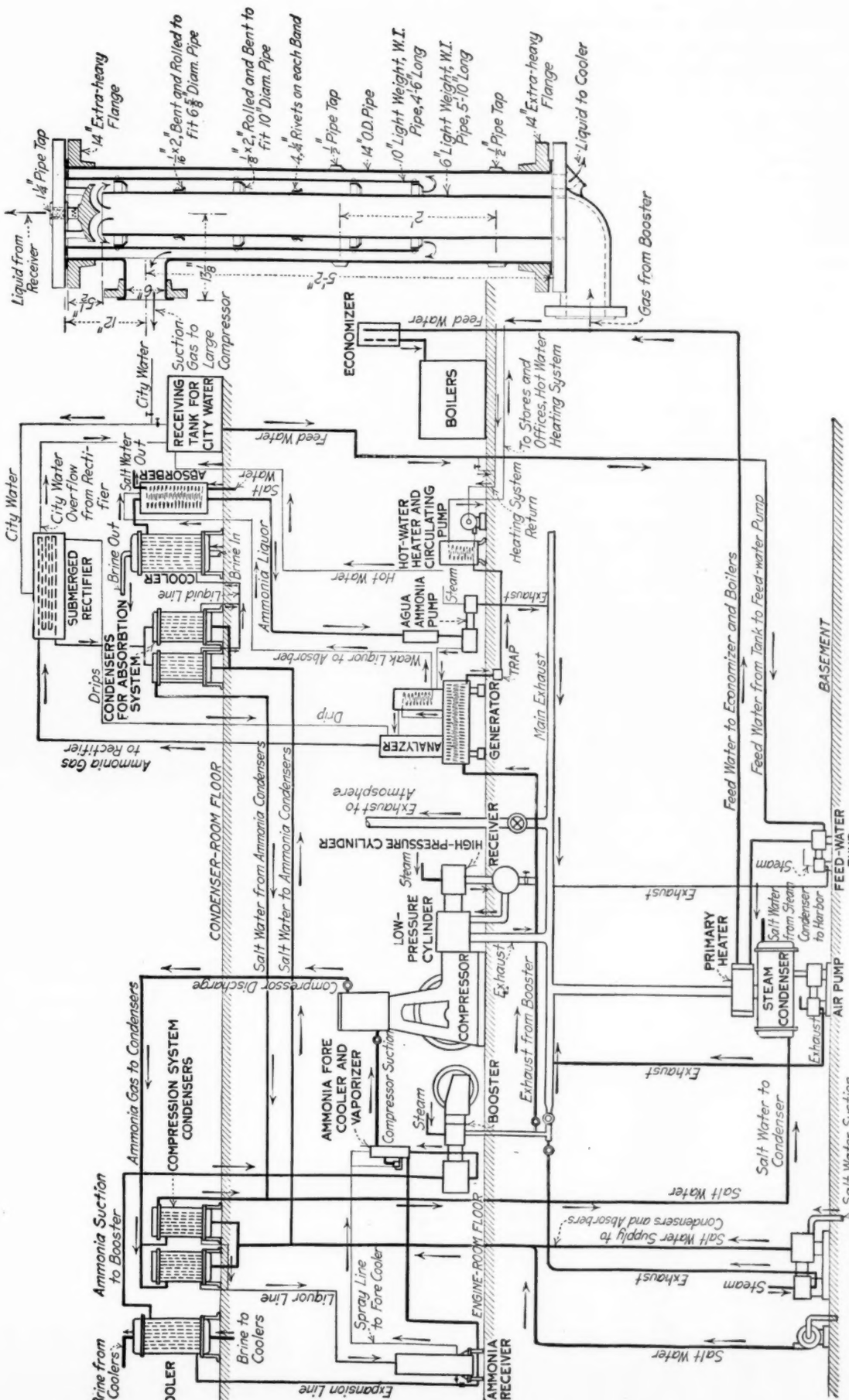


FIG. 2. COMPLETE LAYOUT OF THE BOOSTER AMMONIA COMPRESSOR AS USED WITH COMBINED COMPRESSION AND ABSORPTION SYSTEMS, ALSO SECTION OF AMMONIA FORECOOLER

cylinder per stroke, or 1,680 cu.ft. per min. when running at the usual summer-load speed of 70 r.p.m. At 20 lb. pressure, the weight of gas handled per minute at 70 r.p.m. is 1,680 cu.ft. × 0.0744 = 124.99 lb.

Now suppose the suction vapor is at 45 lb. instead of 20 lb., all other conditions remaining the same. The weight of gas compressed per minute will be 1,680 cu.ft. × 0.1598 = 268.46 lb., or 143.47 lb. in favor of the higher suction pressure.

Look at it another way. Table 1 gives the number of cubic feet of ammonia gas that must be pumped per minute at different suction and condenser pressures to produce 1 ton of refrigeration per 24 hr. With vapor at 6 lb. and a condenser pressure of 184 lb. gage, note that 5.64 cu.ft. of gas is required per minute; whereas with vapor at 30 lb., same condenser pressure, approximately 2.60 cu.ft., 54 per cent. less, is required.

Refer to Fig. 2 and follow the ammonia as it will go through the new system. Gas from the coolers will go to the high-speed booster, which will compress it to between 40 and 60 lb. abs., pumping the gas into the low-pressure gas header. In the discharge line of the booster, between the header and the booster, a special-type separator, Fig. 2, will be placed, as a forecooler. This is perhaps the most interesting part of the system. The discharge gas will run well to the top of the separator, as shown, and be deflected downward by the hood. The device acts as a forecooler and separator for the liquid ammonia from the receiver, all of which will pass through this device on its way to the expansion valves of the coolers, thus bringing the gas, which of course will be superheated, into contact with the liquid, "grabbing" (as

TABLE 1. VOLUME OF GAS PER TON OF REFRIGERATION

Theoretical number of cubic feet of gas to be pumped per minute at different condenser and suction pressures to produce 1 ton of refrigeration per 24 hr.—Sibel.

Temperature of Gas, Deg. F.	Corresponding Suction Pressure, Lb. per Sq.In.	Temperature of Gas, Deg. F. 65	70	75	80	85	90	95	100	105
		Corresponding Condenser Pressure, Lb. per Sq.In. 103	115	127	139	153	168	184	200	218
—27	1	7.22	7.3	7.37	7.46	7.54	7.62	7.70	7.79	7.88
—20	4	5.84	5.9	5.96	6.03	6.09	6.16	6.23	6.30	6.43
—15	6	5.35	5.4	5.46	5.52	5.58	5.64	5.70	5.77	5.83
—10	9	4.66	4.73	4.76	4.81	4.86	4.91	4.97	5.05	5.08
5	13	4.09	4.12	4.17	4.21	4.25	4.30	4.35	4.40	4.44
0	16	3.59	3.63	3.66	3.70	3.74	3.78	3.83	3.87	3.91
5	20	3.20	3,24	3.27	3,30	3.34	3,38	3.41	3.45	3.49
10	24	2.87	2.9	2.93	2.96	2.99	3.02	3.06	3.09	3.12
15	28	2.59	2.61	2.65	2.68	2.71	2.73	2.76	2.80	2.82
20	33	2.31	2.34	2.36	2.38	2.41	2.44	2.46	2.49	2.51
25	39	2.06	2.08	2.10	2.12	2.15	2.17	2.20	2.22	2.24
30	45	1.85	1.87	1.89	1.91	1.93	1.95	1.97	2.00	2.01
35	51	1.70	1.72	1.74	1.76	1.77	1.79	1.81	1.83	1.85

the inventor[3] puts it) the superheat instantly and reducing the gas to saturation point, and also forecooling all liquid anhydrous ammonia for the coolers to the temperature due to the pressure at this point and thereby saving gas compression back through this range, a saving of perhaps 10 per cent. under average conditions. The liquid feed into the separator will be maintained constant by a ball-float trap which will operate a valve in the liquid line from the liquid ammonia receiver to this separator. The liquid, not vaporized, from the bottom of this device will go through the expansion valves to the coolers. The connection from the receiver to the suction-gas main will be taken off the side of the separator, as shown in Fig. 2. The ammonia trap and its valve-operating function will not be in the nature of an experiment, as Mr. Fairbanks has had one working in the Richmond Street plant for the last few years. The large compressor has a maximum speed of 75 r.p.m., but when run in series with the booster, it is the intention to operate at a speed of 60 r.p.m. constantly, as this seems to be the most efficient and economical speed for this machine. The variation in the refrigerating load under these conditions will be handled by regulating the speed of the booster, thereby controlling the capacity of the large machine at a given speed by varying its suction pressure (which is the discharge pressure of the booster) and consequently the weight of gas handled per revolution.

The governor of the booster is designed to control the speed either by the discharge pressure or by an inclosed

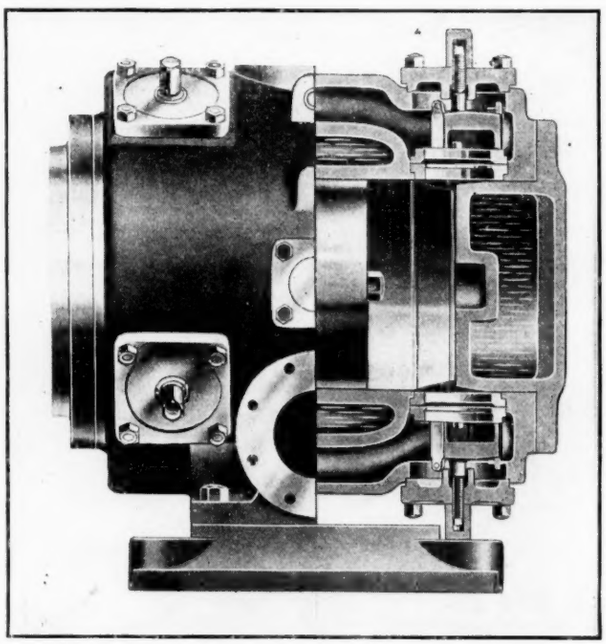


FIG. 3. SECTION OF AMMONIA CYLINDER

flyball-type speed regulator. Actually both of these devices will act as speed limits only and the speed will be varied according to the load, to maintain a constant temperature, by changing the Meyer cutoff, which will be done positively by the operator who handles the expansion valve, thereby linking these two important details together, which is really necessary in carrying temperatures within a range of 1 deg. F., as allowed in the summer.

The steam consumption of the booster will probably run from something like 15 lb. when compounded with the low-pressure cylinder of the 400-ton machine to 30 lb. or more when exhausting into the absorption-machine generator, under which condition its economy is of little consequence for the reason that it is intended to so load the absorption machine in comparison to the compression that it will take all the steam coming to it. It will be easily possible to pump more or less brine through the different coolers to transfer the load, or any part of it, from one machine to the other. This feature of operation is indeed advantageous.

The chief engineer expects quite a little increase in economy because the larger machine can be operated at a fixed constant speed and all regulation done by handling the booster.

The spring tension of the suction valves of the large machine is being changed to meet the new conditions,

[3] F. L. Fairbanks, chief engineer, Quincy Market Cold Storage and Warehouse Co., Boston.

inasmuch as with the high suction pressures it is best to carry a stiffer suction-valve spring, thereby seating the valve quicker and reducing considerably the present slippage on this machine, which is due to the light spring necessary with low suction pressures to get the maximum amount of gas through the valve and into the cylinder at the present suction pressure.

Now follow the course of the exhaust steam from the booster. It will ordinarily pass through the generator of the absorption machine, then on to a large feed-water heater which has a connection for circulating water for heating stores and offices. The water from the heater and from the heating coils returns to a feed tank from which the feed pump delivers it to the boilers. Connection is provided in the booster exhaust line for directing steam to the intermediate steam receiver of the 400-ton compressor, thereby making it possible to compound the booster with the low-pressure cylinder of the 400-ton compressor and to the atmosphere as well as with the absorption-machine generator.

Perhaps the most striking features of the installation are the ammonia valves in the booster. These are of the feather type, flexible and very likely mark the beginning of a departure in ammonia valves and compressor capacities, cylinder dimensions considered, on account of the speeds possible with these valves.

is not an economic loss. To demonstrate this the builders made a series of tests on a compressor so arranged as to permit changing the clearance. The power input to the compressor was measured by carefully calibrated electrical instruments, and the air compressed and delivered was measured by a low-pressure orifice, using Professor Durley's coefficients. The results of these tests are shown in Table 2 and indicate that for clearances ranging from 4 to 12 per cent. the horsepower required to compress a given amount of air is nearly constant. The compression cylinders will seldom have more than 8 or less than 2 per cent. clearance.

TABLE 2. HORSEPOWER PER CUBIC FOOT OF AIR COMPRESSED AND DELIVERED WITH VARYING CLEARANCE

	Clearance, per Cent. of Stroke		
	4.1	8.3	12.4
Duration of test, min.	83	87	91
Average temperature at suction, deg. F.	82.6	73.5	73.1
Barometer, in.	29.33	29.22	29.52
Average speed, r.p.m.	151.52	153.52	154.45
Displacement, cu.ft. per min.	421	426	429
Air compressed and actually delivered, cu.ft. per min.	336	317	302
Volumetric efficiency, per cent.	80.0	74.4	70.5
Horsepower input to motor	47.0	45.18	42.65
Horsepower per cu.ft. of air delivered per min.	0.140	0.142	0.142

On account of its simplicity the valve is low in cost as compared with the usual types of ammonia-compressor valves. The method of compressing ammonia gas, and

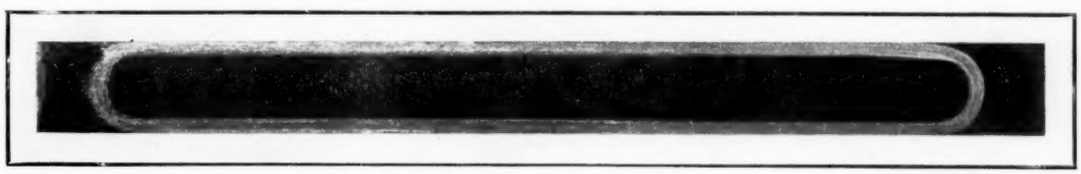


FIG. 4. FEATHER OF VALVE, ACTUAL SIZE, AFTER HARD USE

The valve consists essentially of a ground seat with slots over which rest the feathers or valves, which are nothing more than short strips of flexible ribbon steel. These valves are held in position, but not rigid at any point, by a cover, or guard, with milled grooves for the egress of the gas. The peculiar feature of the feather valve is that it seats by contact and not by impact. That is, the thin strips in bending up from the pressure below them, do so by raising arch fashion. With the discharge stroke finished, the feather or valve seats in much the same way as the hand will when laid on a flat surface and the fingers straightened to flatten the hand. The extremely light weight and flexibility of the feathers reduce inertia and noise to a negligible amount. Fig. 4 is a reproduction of an unretouched photograph of a feather that made more than 40 million closures at 600 to 800 r.p.m.. The bright surface where the valve seated is shown.

The volumetric efficiency of this unit is guaranteed by the builders to be 90 per cent. as shown by measured output.

Engineers have been prone to regard high volumetric efficiency as extremely important from an economic standpoint—have considered that it signified high compression efficiency. Efficiency loss in clearance and slippage is as low as 4 per cent. with this valve. With valves of the voluntary type, such as this feather valve, the reëxpansion of the clearance gas (which, by the way, is the greatest factor bringing about volumetric loss) gives back to the piston practically all the work expended in compressing it: therefore, volumetric loss from this source

particularly its application in this Boston plant, should bring forth discussion of the system's possibilities and limitations. *Power* welcomes such discussion.

The usual type ammonia compressor of 400 tons cost approximately $60 to $75 per ton; the cost of the booster and separator equals $6 to $7 per ton, which shows what saving in first cost per ton is realized by using the booster.

The booster was built by the International Steam Pump Co.'s Laidlaw-Dunn-Gordon plant at Cincinnati under the supervision of Mr. Fairbanks. The new equipment is nearly ready for operation, and the writer hopes soon to present to *Power* readers data of results obtained in service.

Too Many Tests Neglect the Range of Operation and are made, on all machinery including boilers and stokers, at the most efficient point in their operation. Owing to the character of load, boilers have to be run at various rates of steaming, and it is just as important to know the best method of carrying fires at these ratings, night loads for instance, as for the ordinary day load. Draft gages, properly installed and kept in adjustment, can "earn their living" when their indications are intelligently interpreted and the means of controlling the draft are centralized, the instruments and damper controls being so placed that with minimum effort effects of changes in drafts, etc., can be detected immediately. Too many plants have draft gages put in convenient places near the front of the boiler setting, yet the damper controls may be back of, under or on top of the setting, or almost any place but where the instruments are. Control of a damper back of the boiler setting is about like a rheostat control behind a switchboard—the operator having to walk around to the front to note the result of adjustments. And as a consequence, small adjustments are neglected.—Sibley Journal of Engineering.

Philip Carey Boiler Plant

By Thomas Wilson

SYNOPSIS—In this plant, containing 900 hp. in return-tubular boilers and 1,264 hp. in water-tube boilers, about 1,000,000 lb. of steam is generated per day, by 60 tons of coal, at 10½c. per 1,000 lb. of steam and 1.32c. for labor. The coal-handling system is unusually simple, the labor cost per ton for handling averaging 7.15c.

The plant of the Philip Carey Co., Lockland, Ohio, the well-known maker of asbestos products, occupies a site covering 35 acres, on which are 25 buildings mostly of brick and two or three stories high. Exhaust steam in large amounts and some live steam is used throughout the plant for manufacturing processes and for heating, the pressure required varying from 5 to 135 lb. In addition demands for steam are made by three engines aggregating a total of 1,000 i.hp. and by numerous pumps which furnish water throughout the plant and for fire service. All this steam is supplied from a central boiler plant which is operated at a pressure of 135 lb. In 24 hr. this plant generates about 1,000,000 lb. of steam, requiring 60 tons of West Virginia run-of-mine coal costing $1.75 per ton. During the same interval five tons of ashes are removed.

The plant contains six return-tubular boilers, 72 in. by 18 ft., rated at 150 hp. each. A later addition contains one 264-hp. water-tube boiler and two 500-hp. Stirling boilers. All are served by underfeed stokers with automatic control, sufficient draft being afforded by an engine-driven blower. The return-tubular boilers are served by a concrete stack 150 ft. high above the grate and 6 ft. in diameter. The smaller water-tube boiler has an individual steel stack 80 ft. high and 48 in. diameter, while the two 500-hp. Stirling boilers are served by a steel stack 78 in. diameter and 149 ft. high. All the water-tube boilers are maintained in operation night

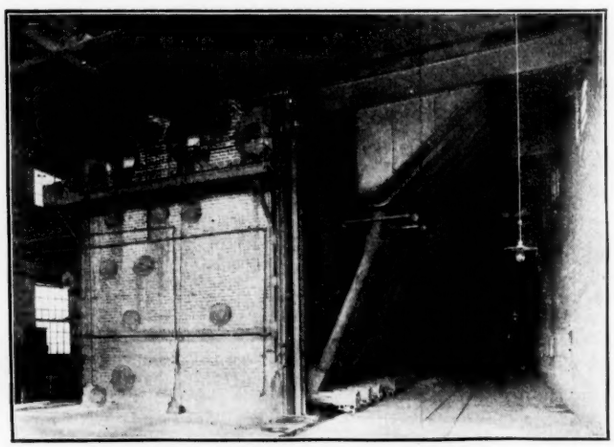


FIG. 1. BOILER ROOM OF PHILIP CAREY CO.

and day and as many of the return-tubular boilers as are needed to meet the demand for steam.

Much of the feed water is condensation and returns from the mills. Some makeup, however, is required, and this is first passed through a combination water softener and filter, using lime and soda ash. The fresh water contains as high as 28 grains of incrusting solids per gallon, but in the treatment this is reduced to 6 grains and an alkalinity of 1½ per cent. at a cost of 1c. per 1,000 gal. From the softener the water flows by gravity to either of two open feed-water heaters, and with the condensation is returned to the boilers at a temperature averaging around 208 deg. F.

The topography of the land in the vicinity of the boiler house permitted installing an unusually simple coal-handling system. The boiler house is built in a depression and the railway track on which the coal is delivered is carried out from the general level on a trestle which is 18 ft. above the boiler-room floor. As is evident in Fig. 2, this trestle passes over an open concrete storage bunker into which the coal is dumped directly from the cars. At the bottom the bunker has a number of gates

FIG. 2. EXTERIOR OF PLANT, SHOWING COAL TRESTLE

through which the coal may be discharged into an industrial car. Sufficient headroom has been left under the bunker for a man to operate this car, and a narrow-gage track at the boiler-room level extends from one end to the other.

From the coal bunker the car is shoved onto a hydraulic elevator having a 14-in. casing. It is operated by the water pressure in the main supplying the plant, which averages about 140 lb. The elevator carries the car full of coal to a level above the bunkers and as there is a transfer truck underneath, the car is pushed directly onto the track running over the steel bunkers supplying the boilers. There are four of these bunkers, and the car, which is of the side-dump type, may be emptied into any one desired. The elevator descends by its own weight, the 14-in. casing discharging into a 3-in. line having a valve which may be manipulated as necessary to limit the speed upon the return trip.

The ashes are removed in much the same way. A track extends in front of the boilers so that the car may be pushed to any point desired. It is then raised by the elevator to the track above the bunkers, which extends out of the end of the boiler room over two ash tanks. One of these tanks discharges into wagons backed underneath, while the second tank discharges directly into the railway cars bringing in the coal.

The plant is operated day and night on two shifts, one man on each shift handling all the coal and removing the ashes. In addition the night man helps about the boiler room, and if the two bunkers serving the water-tube boilers had been made a trifle larger, his services could have been dispensed with as far as the coal and ashes are concerned. The two men handle a total of 60 tons of coal and 5 tons of ashes. Their wages per

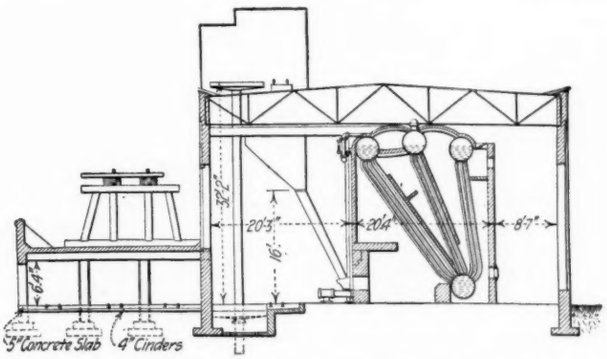


FIG. 3. SECTIONAL VIEW THROUGH BOILER ROOM

day amount to $4.65, which, divided by 65, gives a labor charge of 7.15c. per ton. With the elimination of a night man, this labor charge would have been reduced to 3.57c. per ton, which is unusually low. The cost of the coal-handling apparatus from the railway car to the stoker hopper totals $15,340. This includes the open concrete bunker, the trestlework and track, the industrial car and track, the elevator, the steel coal bunkers in the boiler room and the ash trestles and tanks previously mentioned. Including interest, depreciation and taxes, the overhead at 12½ per cent. would amount to $5.25 per day, which reduces to 8c. per ton of coal or ashes handled. The cost to operate the elevator has not been computed, but it amounts to a small item per ton of material handled. Maintenance items would also be small owing to the simplicity of the outfit. With the two men employed, the overhead and labor per ton totals 15.15c., and with one man employed, which the extension of two of the bunkers would make possible, the cost would be only 11.57c. per ton.

In the boiler room two firemen and one coal handler are employed on each shift, making a total of six men for the 24 hr. Their wage per day totals $12.65. The sixty tons of coal costs $105. As nearly as can be estimated, 1,000,000 lb. of steam is generated per day. This would mean an evaporation of something over 8 lb. of water per pound of coal, which is not unreasonable when it is considered that on test the water-tube boilers showed an evaporation of over 11 lb., and the fire-tube boilers an evaporation of 9½ lb. The load is adjusted so that the various units operate at high efficiency.

On the basis given, the coal per 1,000 lb. of steam would cost 10½c. and the labor 1.32c. These figures are lower than usually obtained in the average power plant. The credit is largely due to Max Paul, who has been chief engineer for the Carey company for twenty years. W. J. Moeller is general superintendent and S A. Coffing, engineer of construction.

PRINCIPAL EQUIPMENT OF PHILIP CAREY BOILER HOUSE

No.	Equipment	Kind	Size	Use	Operating Conditions	Maker
6	Boilers	Return-tubular	72-in. x18 ft.	Generate steam	135 lb. pres., underfeed stokers	McIlvain & Spiegel Boiler and Tank Co.
1	Boiler	Hor. water-tube	264-hp.	Generate steam	135 lb. pres., underfeed stokers	Babcock & Wilcox Co.
2	Boilers	Stirling	500-hp.	Generate steam	135 lb. pres., underfeed stokers	Babcock & Wilcox Co.
15	Stokers	Underfeed		Serve boilers	Provided with engine-driven blower and automatics	The Under-Feed Stoker Co. of Am.
2	Pumps	Duplex	12x8x18-in.	Boiler feed water		Canton-Hughes Pump Co.
1	Pump	Duplex	10x6x10-in.	Boiler feed water		Henry R. Worthington
1	Pump	Duplex	10x6x12-in.	Boiler feed water		Stilwell-Bierce & Smith-Vaile
1	Heater	Open	2,000-hp.	Heat feed water	Water temperature, 208 deg. F.	Pittsburgh Feed Water Heater Co.
1	Heater	Open	1,000-hp.	Heat feed water	Water temperature, 208 deg. F.	Bates Machine Co.
1	Water Softener	Cold process	10,700 gal. per hr.	Soften feed water	Lime and soda ash, filter	Dodge Manufacturing Co.
1	Car	Industrial	1½-ton	Handle coal and ashes		C. W. Hunt Co.
1	Elevator	Hydraulic	2-ton	Elevate coal	Use pressure from water mains	Curran Elevator Co.

The Reactance Was High

By E. C. Parham

The torque of an induction motor decreases rapidly when the voltage applied to the motor is decreased, therefore a regulating resistance or a regulating reactance connected into the supply lines does not afford a very satisfactory means of varying the speed of the rotor. So great is the effect of impedance in the supply lines of an induction motor that instances can be cited where unsatisfactory operation has been converted into satisfactory operation, on temporary installations, simply by removing the supply wires from their insulators and bunching them together so that mutual induction could overcome the self-induction and thereby eliminate the choke-coil effect of the self-induction.

A construction company complained that a 10-hp. 220-volt three-phase induction motor that was belt-connected to a circular saw was unable to do its work, although smaller motors were doing the same work on other jobs.

An inspector was sent to check the layout. He immediately noticed that the connecting belt was very short and that it would have run off had the belt guards not prevented it. The belt was cut, drawn together and relaced, but the tightening apparently helped matters only when the saw was not cutting. The saw would stick as soon as it got well into the work, and this would slow the motor to a critical speed at which again there was slipping of the belt. The controller contacts and the resistance connections were then checked to insure that the resistance actually was being cut out when it was supposed to be; everything in the rotor circuit proved to be regular.

On applying the voltmeter to the supply lines, however, the voltage actually applied to the motor was found to be 180 volts with the saw running, but not cutting; as soon as the saw was pressed into the cut, the voltage would drop to as low as 140 volts. Apparently the supply lines from the service transformer were neither long enough nor small enough to account for the excessive voltage drop, and they were bunched together for convenience in handling, so their reactance was at a minimum. The voltage of the secondary of the supply transformer was then checked right at the transformer and was found to be 240 volts, 20 volts above the voltage rating of the motor. From the transformer the lines went to a little construction shanty where the switch, the fuses and the meter were

located. Where the motor wires came out of the other side of the shanty, instead of running directly to the motor each of them was connected to one end of about 600 ft. of wire that was wound on a reel, the other ends going to the motor. The reactance of these coils was what reduced the voltage at the motor terminals and caused it to slow down.

It developed that the motor frequently had to be shifted, according to the location of the work. In order to avoid the trouble and delay of lengthening the lines each time that the motor had to be shifted, the foreman had conceived the ingenious idea of winding the lines onto the reels, which could be turned on their centers whenever it became necessary to lengthen the lines.

Indicator-Diagram Peculiarities

By Thomas J. Rogers

SYNOPSIS—An explanation of indicator diagrams showing improper action of the valves, restricted exhaust area, peculiarities of expansion lines, reopening of the steam valve and eccentric out of place.

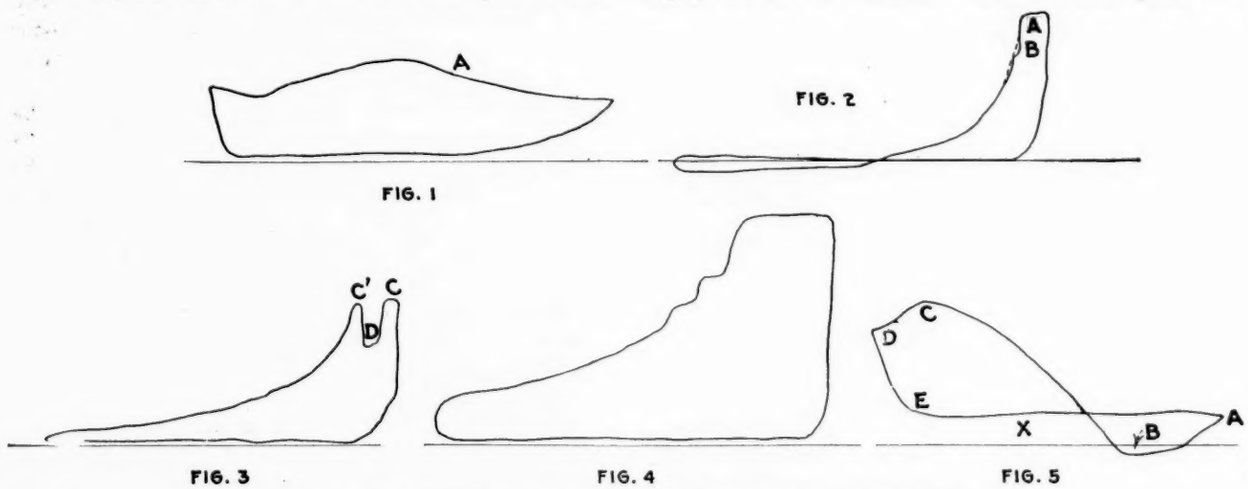
The application of an indicator to the cylinder of a steam engine frequently reveals curious conditions. The diagram, Fig. 1, was taken from an 18½x48-in. engine.

At the time the diagram was taken, the engine was developing 60 hp. and exhausted 48½ lb. of steam per horsepower-hour. How much water was exhausted is not known, but there was so much in the cylinder that it was scarcely safe to keep the cylinder drip-cocks closed while indicating. Probably not less than 7 lb. of coal per horsepower-hour was used—about a ton a day more than should have been required. As the engine had been running 10 years in this condition, it had wasted approximately $15,000 worth of coal.

The engine had a main valve, and a cutoff valve placed outside the steam chest, the latter being controlled by the governor. The diagram shows satisfactory compression to the point where the main valve opens the steam ports, when the pressure, instead of rising to nearly boiler pressure, drops until the piston has advanced about one-eighth stroke. At this point the cutoff valve on the outside of the steam chest slowly opens until the maximum pressure is realized at the highest point shown at the point of cutoff at the beginning of the expansion line *A*. The opening of the cutoff valve is so late that cutoff, which should have taken place at less than one-quarter, occurs at one-half stroke. Then the volume of steam in the steam chest, together with that in the cylinder, is expanded to about 40 lb. absolute, instead of to 20 lb. or less as it would have been under favorable conditions. The exhaust opens late and the counter-pressure falls slowly, the back pressure being 5 lb., when it should not have been more than ½ lb.

The diagram, Fig. 2, shows in the expansion line peculiarities that are sometimes taken as representing improper action of the indicator. The expansion line from *A* to *B* is substantially vertical as is only found where the clearance is small. In such instances the pressure in the cylinder falls rapidly directly after cutoff, because a small movement of the piston will serve to largely increase the volume of steam in the cylinder as compared with the volume at cutoff, and correspondingly decrease the pressure. The steam being cut off suddenly at *A*, the piston traveling at a mean speed of between 500 and 600 ft. per min., the horizontal distance to the point *A* representing less than 1-in. travel of the piston, the pressure under the indicator piston is almost instantly materially reduced. The spring being under considerable tension starts the indicator piston down, and the momentum that it acquires carries the pencil ahead of the position due to the steam pressure, as in diagrams from high-speed engines or in those taken with a light spring, and the pencil is found alternately too low and too high. In such instances the pencil will fall below its correct position, then the reaction will carry it above.

In this instance the first evidence of reaction is at *B*, where the reciprocating parts of the indicator recover from the effects of their downward impetus. Had this occurred later, the pencil would have been carried up considerably out of its true course, but at this early



FIGS. 1 TO 5. INDICATOR DIAGRAM HAVING PECULIAR FEATURES

point of the stroke the pressure is falling too rapidly for this to occur, so that the reaction amounts to scarcely more than a pause in the downward motion. Below *B* a short distance there is another pause, after which the expansion line gets into proper shape, being assisted by the more gradual fall in pressure as the volume of steam is increased. Had the same thing occurred later in the stroke, as from a cutoff when the volume of steam in the cylinder was twice or three times as great, the result would have been the wavy expansion line frequently seen.

There is still another cause tending to produce a straight line from *A* to *B*. The actual curve of the diagram is usually below the theoretical curve directly after cutoff, especially where full steam is worked for a fraction of the stroke only, from the fact that condensation is taking place with great rapidity to supply the heat lost by the exposed surfaces during expansion and exhaust; so that taking into consideration both these causes, each operating in the same direction, the position of the line from *A* to *B* is accounted for without attributing any fault to the indicator. Finally, drawing the theoretical curve from the point of cutoff, the variation is not so great as a glance at the diagram would seem to indicate. A part of this curve is drawn in dotted line and shows the variation a little greater than it actually is.

Some Additional Peculiarities

The diagram in Fig. 3 shows peculiarities that cannot be explained in this way. The initial pressure was but one pound less than boiler pressure, and the valve closed sharply at *C*, after the piston had made about 5 per cent. of its stroke. The effort to recover after the usual somewhat too rapid fall of the pencil is seen at *D*, and from some cause the pressure rises again to nearly the initial pressure, making apparently two points, *C* and *C′*, of cutoff. The cause of this was that the cutoff valve, which dropped in closing, afterward left its seat, and from the amplitude of the steam ports actually admitted steam of nearly boiler pressure the second time. From a short distance below *C′* the expansion line falls too slowly for some distance, indicating that the valve leaked while seating.

The engine from which this diagram was taken was a small one, and the aggregate length of the ports controlled by the cutoff valve was equal to about twice the diameter of the cylinder. With this length of port the cutoff proper occurred when the valve was dropping at its highest velocity, and through only a short distance. The effect of this was that the pressure in the cylinder was not reduced enough while the valve was closing, so that the unbalanced pressure in the steam chest would hold the valve to its seat against the jar of the air cushion, which stopped it.

Two remedies suggested themselves—one was to provide for closing the valve slower, the other to arrange a guard to prevent the valve leaving the seat horizontally. The latter remedy was used successfully.

Trouble similar to this, though not so marked in extent, often occurs in four-valve engines. The result generally appears somewhat later in the stroke in the form of an irregularity in the expansion line, as shown in Fig. 4, and high terminal pressure. This is usually explained as being the result of wet steam, a small cylinder, or in any but the correct way. The explanation is, I believe, that the valve does not close properly, or reopens a little and does not get properly seated until the piston is some distance beyond where cutoff apparently takes place.

In Fig. 5 it would at first appear that the admission line is at the left and that the cutoff is at *C*. A little study will show that the admission end of the diagram, is at the right. If the admission was at the left, then *DC* is the admission line and *CB* the expansion curve. It would be possible by bringing about peculiar leaks to get either of these lines, but it is improbable that leaks occurring in the ordinary way should be responsible for changing the direction of flexure of either of the curves as shown.

Beginning at the right end of the diagram, the reading is plain. At *A* the piston begins to move ahead, but the valve being behind time, the admission port is covered. There is a pressure of 26 lb. absolute in the clearance space. With the piston moving and the port remaining covered, the pressure fails by expansion to *B*. At this point, about one-quarter stroke, the valve begins to open the port, but the motion of the piston is so rapid and the space to be filled with steam so large that the pressure increases slowly, as represented by the line *BC*. It is in respect to this line an aggravated case of late admission without assistance from compression. Under these conditions the upward flexure of the line *BC* is accounted for and is what would be expected.

At *C* steam is cut off; *CD* is the expansion line. The exhaust opens late at *D* and the pressure falls slowly, until beyond *E* it is equal to that against which the piston is working. The line from *D* by *E* to *A* is the line of counter-pressure, there being no compression.

The engine from which this diagram was taken is 22 in. diameter cylinder by 30 in. stroke, running at 75 r.p.m. It kept running, doing its regular work with the misplaced eccentric, but pounded so badly the indicator was applied to ascertain the cause. The exhaust was against the high pressure shown, because the exhaust steam was used for other purposes requiring that pressure. But for this use of the exhaust steam the trouble would have manifested itself at the coal pile.

What! Again?

Perpetual Motion Discovered

J. H. Gaither and W. J. Harvey, of Wood River, have completed a model of perpetual movement. They have worked on the plans of this model over five years with hard and constant study. This piece of mechanism is made entirely of cypress lumber and shows mechanical skill throughout. The machine will develop one-half horsepower it is expected, and the main flywheel has a speed of 100 revolutions per minute of constant speed. They have taken the matter up with a well-known lawyer of Washington, D. C., to secure patent rights, which they expect in the near future. This patent it is said, if approved, would be valued at $200,000.

This movement, complete, measures 38½ in. long, 24 in. wide and 24 in. high.

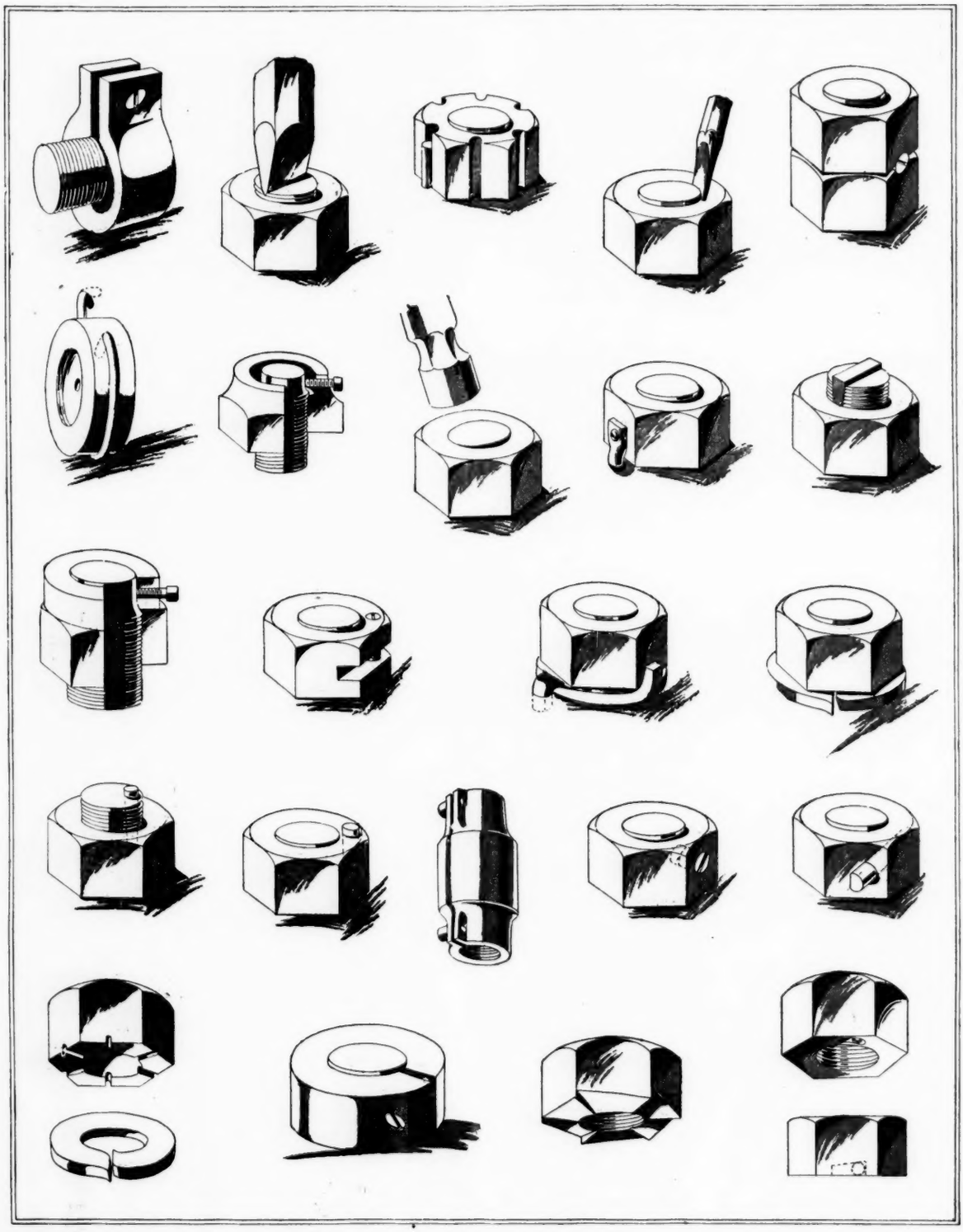
Mr. Harvey is an expert automobile repairman and a brother-in-law to T. J. Hart, of Hartford station.

Mr. Gaither is a telegraph operator at Hartford station. Both lived in Wood River until six months ago. This model of perpetual movement was being worked on at Hartford, but is now in a shop at Wood River.—Alton (Ill.) "Telegraph."

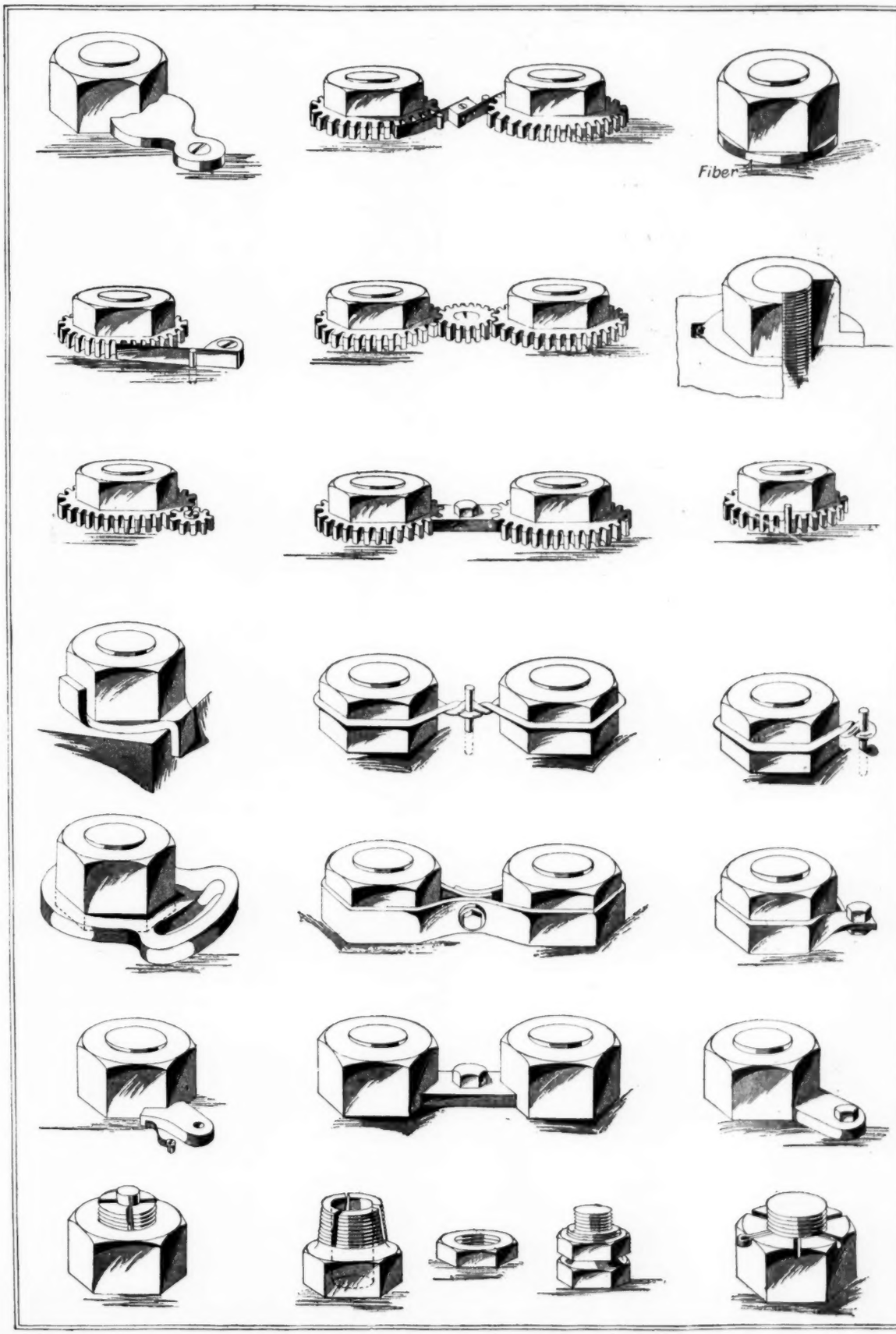
The Three Kinds of Motion are uniform, accelerated and retarded. At uniform motion a body moves over equal spaces in equal time. Accelerated motion is that by which the velocity increases while the body is moving. Retarded motion is that by which the velocity decreases while the body is moving.

From a Small-Shop Notebook

By John H. Van Deventer



A WIDE VARIETY OF LOCKNUTS AND LOCKING DEVICES FOUND TO



MEET PRACTICAL REQUIREMENTS IN VARIOUS CLASSES OF WORK

(Two pages of similar devices will appear next week)

—From American Machinist

Ventilating Electric Machines

By Gordon Fox

SYNOPSIS—The author explains the cause of heating in the various parts of electrical machinery, how heat is dissipated from these parts and the different methods of ventilation employed.

The function of electrical machinery is to convert mechanical energy into electrical, or vice versa. Losses are inevitably connected with any such conversion, the lost energy appearing in the form of heat, and they are objectionable for two reasons: They lower efficiencies, and they increase the temperature of the machine. Losses in electrical machines may be classed as electrical losses due to the resistance of the conductors through which the current flows, magnetic losses due to hysteresis, and mechanical losses such as windage, bearing and brush friction. Copper losses raise the temperature of the conductors, iron losses heat up the cores, sparking and brush friction heat up the commutator.

Heat may be transferred in three ways—by radiation, convection and conduction. Radiation occurs when heat is given off from hot bodies in rays which radiate in straight lines in all directions, as heat given off from the surface of a mass of hot metal. Convection is heat transferred from one point to another by currents, as in the heating of water in a boiler, or as a room is heated by currents of warm air. Conduction is transfer of heat through materials in a manner similar to the flow of an electric current.

Heat Dissipated in Electrical Machines

The difference of temperature between two points is similar to potential difference in an electric circuit. Thermal and electrical resistances are analogous, and the flow of heat by conduction follows laws similar to those governing the flow of electric current. Electrical machines dissipate heat by all the foregoing means. Conduction is of primary importance in the transfer of heat from the point of origin to the point of dissipation; radiation and convection then complete the work, the last named generally playing the important part in carrying the heat away from the exposed surfaces.

There is always a flow of heat from a point of higher to a point of lower temperature. The extent of the flow is a function of the quantity of heat generated, the length and section of the paths of flow and the heat conductivity of the intervening material. Materials differ in their conductivities for heat in much the same way that they differ in electrical conductivities. A material having low heat conductivity involves high temperature difference per unit of distance and is said to have a steep heat gradient, where a good conductor of heat has a low heat gradient. Copper is an excellent conductor, and iron a medium conductor, of heat as well as of electricity. Insulating materials are always poor conductors of heat. Laminated iron is not a very good heat conductor in the direction across the laminations. The heat gradient is about ten times as steep across laminations as through solid material.

When there is no load upon an electrical machine the losses are, for the most part, iron losses. These iron losses comprise hysteresis losses incident to the repeated reversal of the magnetism or realignment of the molecules, and eddy-current losses due to electric-current flow through local circuits in the iron. Hysteresis losses are restricted by using high grades of iron. Eddy-current losses are reduced by laminating the iron and insulating the laminations from each other. This insulation should evidently be sufficient to prevent eddy currents, but should not be excessive because of the resistance to heat flow thus introduced and the increased dimensions of the core for a given volume of iron. When a machine is under load, the copper losses are in evidence and vary with the load. The working current is not alone contributory to copper loss; the local or eddy currents may be of considerable value, particularly in large conductors. Eddy-current losses in the copper conductors are restricted by the use of laminated conductors and by transpositions that cause locally induced voltages to neutralize each other.

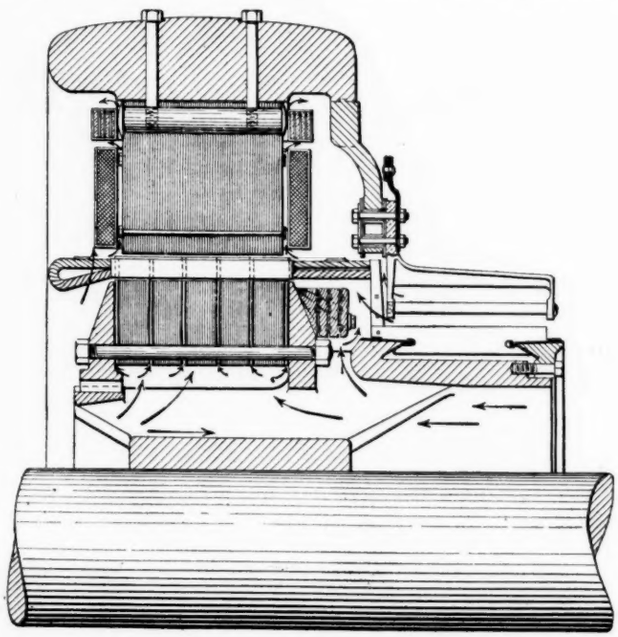


FIG. 1. SHOWS THE METHOD OF VENTILATING SLOW-SPEED ELECTRICAL MACHINES

Consider the conditions existing in the armature winding of a direct-current generator under load. Heat is generated in the armature teeth and in the core, due to iron losses. These losses will be more pronounced in the teeth, owing to the higher magnetic density. Heat will be liberated in the conductors, due to copper losses. Portions of these conductors are buried in the slots, surrounded by heated iron; other portions, the end windings, are fairly exposed to the air. Evidently one path of heat flow will be from the central portions of the conductors to the end windings by conduction, thence by convection to the atmosphere. A certain heat gradient is required to set up this flow, its slope being low through the copper, high across the insulation and considerable from the outside surface of the insulation to the air. The steeper this gradient the more rapid the flow.

There are other paths of heat flow within an armature. This illustration is introduced simply to make evident the fact that temperature differences must exist, and there is a tendency for local hot spots to occur, more particularly in machines having large cores or otherwise involving working material far removed from external surfaces.

The temperatures in electrical apparatus are of prime importance because of their influence upon insulating materials. Were copper and iron alone involved, the allowable temperatures might be much higher than they are, but insulation is of equal importance throughout an electrical machine. It is thus of little benefit to have one portion of a machine cool if at another point a damaging temperature is reached. The temperatures must be equalized so far as possible and the piling up of heat in spots avoided. To accomplish this it is necessary to provide means for the dissipation of heat at points near its origin. Convection must be depended upon for this purpose. Water cooling is not feasible in rotating electrical machines; air is the only practical agency for carrying off the heat. Convection can be secured by directing air past the surfaces of a stationary part—for example, the cooling of field coils by the fanning action of armature rotation. Cooling may be obtained by movement of a body through the air, as in a revolving armature. Cooling may be provided by forcing air under pressure through definite paths and channels. Most machines combine all these methods, but in different types different features predominate.

Methods Employed for Air Circulation

There is found to be a definite relation between the amount of heat to be dissipated in a machine, the quantity of air moved past its surfaces and the resulting temperature rise. One kilowatt of energy will raise the temperature of 100 cu.ft. of air about 18 deg. C. in one minute. With a given loss and a definite allowable temperature rise a definite volume of air must be moved. This air must be circulated through a machine in such a manner as to reach all ventilating surfaces and prevent the formation of hot spots or zones. Thus, in an armature air may be brought along the shaft through an open spider, thence out through ducts in the laminations into the air gap, as shown in Fig. 1. Dividing the stacking into sections in this manner makes it unnecessary for the heat to travel long distances across the laminations and thus prevents heat accumulations. The field coils of an electrical machine naturally tend to retain their heat, comprising, as they do, many turns of insulated wire. Field coils on the larger machines are usually arranged to be spaced out from the iron core to allow the passage of air under them, as shown in the illustration. Not infrequently they are sectionalized, and spacing strips are inserted to give the air access to interior surfaces.

In the earlier days it was the practice to use solid risers and to cover the end windings of an armature with hoods to prevent the entrance of dirt. This practice has been entirely abandoned for open risers in all but the smallest machines, and coil ends are always exposed. In early practice the use of paddles or fan blades to circulate the air was considered as a remedy for faulty designs. At present many machines of the best design are so equipped. The windage of the machine is increased a trifle, but the gain in rating, the cooler and more efficient operation more than offset the slight loss. Ventilating provisions have been applied, not only to the interior parts of machines, but to the exteriors as well. Most alternators and induction motors are now built with so-called "skeleton" frames, which expose the stator iron to the air and thus aid heat dissipation.

The amount of ventilation required in a machine depends materially upon its speed. A large, slow-speed machine has a small output per pound of material and a small output per square foot of radiating surface. The losses allowable are limited by the efficiency reduction. These comparatively small losses can be easily dissipated from so large a surface. As the speed of a machine increases, its rating increases and its output and losses

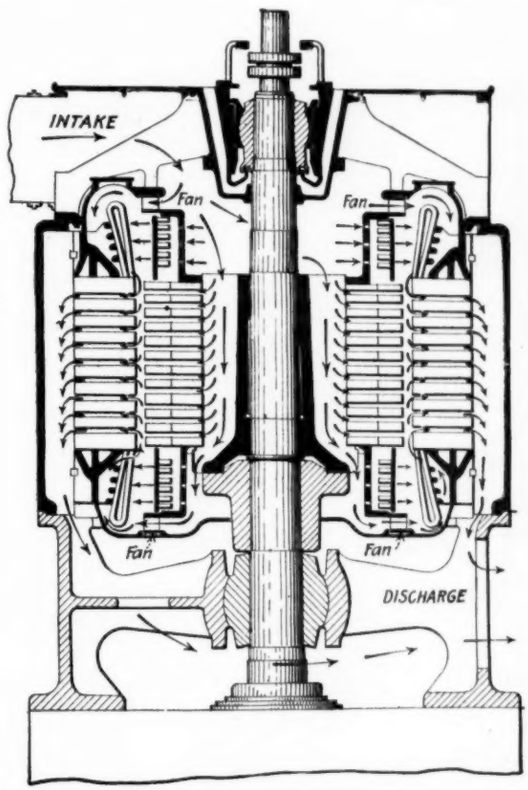


FIG. 2. REPRESENTS THE AIR CURRENTS THROUGH A HIGH-SPEED TURBO-ALTERNATOR

are both enlarged. It must then dissipate more energy from the same surface. The losses in a large, slow-speed frame are limited by efficiency reduction; in a small, high-speed frame dissipating ability becomes increasingly important. The turbine generator represents the extreme in high-speed machines, and proper ventilation is one of the most difficult features of its design.

Forced cooling has been carried to its most complete application in the turbo-generator. Here the ordinary open ventilation would be not nearly ample, particularly for reaching interior points. The machines are therefore inclosed, and tremendous volumes of air are circulated through channels and ducts specially provided. This circulation may be obtained from self-contained fans on the rotor shaft or by the use of separate blowers. Fig 2 shows a vertical steam-turbine driven alternator ventilated by fans on each end of the rotor; the arrows indicate the direction of the air currents. In large units

the separate blowers have the advantages of greater efficiency and control of the volume of air to correspond with load changes. The air may be taken from and exhausted directly into the engine room. In large units the amount of heat thus liberated may be objectionable, and it becomes necessary to connect the intake and exhaust outside the engine room. Forced ventilation on a large scale may involve such quantities of air that a small percentage of dust may mean considerable deposit in the ventilating ducts. Therefore, in large installations it is becoming the practice to wash the intake air by passing it through a water spray.

The influence of ventilation is immediately evident when a machine is inclosed for protection. The rating of the machine is immediately reduced and its temperature increased. This effect is counteracted in some machines by a fan on the rotor or armature shaft; inlets and outlets are provided in the frame, which may be connected outside the room. Forced circulation then supplants the natural circulation of an open machine.

With improvements in design to overcome commutation and other difficulties, ratings are placed largely upon a heating basis; and ventilation thus becomes a matter of increasing importance.

Circular Mil Formula Derived

SYNOPSIS—How the formula is derived for finding the circular mil cross-section of conductors, and how this formula may be transposed to obtain other formulas. Examples are given and worked out, showing the use of the various formulas.

The derivation of the formula, $\text{circ.mils} = \frac{21.6DI}{E_d}$, for finding the sizes of conductors should prove of interest to those not already familiar with it. With the exception of the Ohm's Law formula, there are no others used to the same extent as are those for finding the sizes of conductors.

Before the size of a conductor can be found, certain elements must be known, namely: The current in amperes that is to be transmitted (I), the allowable volts drop in the line (E_d) and the length of the line one way in feet (D).

In measuring the cross-section of wires, the unit circular mil is used and is equal to the cross-sectional area of a circle that measures just 0.001 in. diameter. The use of this unit for measuring the areas of wires has several advantages. Chief among them is that it gives a convenient value for the areas. For example, if the area of a No. 10 B. & S. conductor is expressed in square inches, it will give a value 0.00000766; in circular measure 9.89 circ.mils. It is obvious that the value 9.89 circ.mils is a more convenient expression than 0.00000766 sq.in. Again, the area of a round conductor in circular mils is equal to its diameter in mils squared. To illustrate: The diameter of a No. 10 B. & S. wire is 101.89 mils; the square of this, or $101.89 \times 101.89 = 10{,}381$ circ.mils. The inverse of the foregoing will be true.

A circular-mil foot is a wire 1 ft. long and 1 circ.mil in cross-section. The resistance of 1 circ.mil-ft. of copper at 75 deg. F. is 10.8 ohms.

Assume a conductor of a length represented by L. Its cross-section may be considered made up of wires 1 circ.mil in cross-section. A 1-ft. length of one of these wires, which the cross-section of the conductor is considered made up of, will have a resistance of 10.8 ohms (the resistance of 1 mil-ft.). According to the law for joint resistance of parallel circuits, if two elements of equal resistance are connected in parallel, the joint resistance will be one-half that of one; if three are connected in parallel, one-third that of one, etc. Since 1 circ.mil-ft. has a resistance of 10.8 ohms, if two of the units are connected in parallel, the joint resistance will be one-half of $10.8 = 5.4$ ohms. Connecting ten in parallel, the joint resistance will be one-tenth of $10.8 = 1.08$ ohms. From this it is seen that the resistance of 1 ft. of copper wire will equal the resistance of 1 mil-ft. (10.8 ohms) divided by the number of mil-feet in parallel, or in other words divided by the circular mil cross-section of the conductor and may be expressed, the resistance of one foot of copper at 75 deg. F. is $\frac{10.8}{\text{circ.mils}}$ ohms.

When a number of equal resistances are connected in series, the total resistance of the circuit equals that of one resistance times the number of elements in series. The conductor selected may be considered made up of L number of one-foot sections in series. Therefore the resistance (R) of the conductor will equal the resistance of one foot $\left(\frac{10.8}{\text{circ.mils}}\right)$ times the total length in feet (L), from which the resistance of any conductor may be found by the formula,

$$R = \frac{10.8}{circ.mils} \times L = \frac{10.8L}{circ.mils}$$

At the start it was not intended to derive a formula for finding resistance, but instead the circ.mils cross-section. However, the resistance formula contains the circular mils, therefore it will be a simple matter to transpose the elements to find the circular mils. In this case transposing R and circ.mils will be the correct thing to do; then the formula will read $\text{circ.mils} = \frac{10.8L}{R}$.

The length of the circuit is not expressed in the total length of conductor, but the distance one way. In this case if the distance in feet one way is expressed by D, then the total length in feet (L) of conductor in a two-wire circuit will equal ($2D$). Hence $L = 2D$. As the circular-mil formula now stands, it will be necessary to calculate the allowable resistance of the conductors before the circular mils may be found; consequently, if some element that is known can be substituted for R, it would be desirable. The resistance (R) of any circuit equals the volts drop (E_d) across the circuit divided by the current (I). Hence, $R = \frac{E_d}{I}$. By substituting $2D$ for L, and $\frac{E_d}{I}$ for R in the formula $\text{circ.mils} = \frac{10.8L}{R}$, it will read $\text{circ.mils} = \frac{10.8 \times 2D}{\left(\frac{E_d}{I}\right)} = 10.8 \times 2D \times \frac{I}{E_d}$ from

which circ.mils $= \frac{21.6DI}{E_d}$. By transposing circ.mils and E_d, a formula for finding the allowable drop in the line is derived, which is $E_d = \frac{21.6DI}{\text{circ.mils}}$.

The expression may also be made to read $D = \frac{\text{circ.mils} \times E_d}{21.6I}$ for finding the length of the circuit in feet one way, and $I = \frac{\text{circ.mils} \times E_d}{21.6D}$ for determining the current that can be transmitted in a known circuit for a given voltage drop.

The following examples will illustrate the use of the foregoing formulas:

A load located 750 ft. from the generator requires 80 amp. What size wire is required allowing a 5-volt drop in the line?

$$Circ.mils = \frac{21.6DI}{E_d} = \frac{21.6 \times 750 \times 80}{5} = 259{,}200\ circ.mils$$

A 250,000-circ.mil conductor will be large enough to take care of this load. The next size above this is 300,000. In calculating the size of conductors, the Board of Fire Underwriters' ratings must be kept in mind, and a conductor smaller than allowed by this table should not be installed. In this case the current is 80 amp., and the Board of Fire Underwriters' table for rubber insulation calls for a No. 3 B. & S. conductor, which has a cross-sectional area of 52,630 circ.mils, or about one-tenth the size calculated. As far as they are concerned, this size of conductor may be installed, but the voltage drop in the line would be excessive, as may be proved by using the formula,

$$E_d = \frac{21.6DI}{circ.mils} = \frac{21.6 \times 750 \times 80}{52{,}630} = 24.6\ volts$$

The distance 80 amp. may be transmitted over a No. 3 B. & S. wire, with a 5-volt drop, can be obtained by the expression,

$$D = \frac{circ.mils \times E_d}{21.6I} = \frac{52{,}630 \times 5}{21.6 \times 80} = 152\ feet\ approx.$$

The current that may be transmitted over a No. 3 B. & S. feeder 750 ft. long, allowing 5-volts drop, is found by the formula,

$$I = \frac{circ.mils \times E_d}{21.6D} = \frac{52.630 \times 5}{21.6 \times 750} = 16.2\ amperes$$

"Simplex" and "Power" Turbo Blowers

One way to make boiler draft with a steam jet is to blow air through it like the McClave and other blowers. Another way is to discharge the steam jets into a turbine instead of directly into the air and let the turbine run a fan to force the air under the furnace grates. The Power Turbo-Blower Co., 17 Battery Place, New York City, does it the latter way. A description of their first blower was published on page 426, Vol. 31, 1909.

This company now makes two designs of turbo-blowers, the "Simplex" and the "Power," for medium and small-sized boilers and for large boiler units respectively. Both are improvements over the original blower.

The "Simplex" blower, Fig. 1, is made with but few parts, and its construction has been simplified by making the turbine casing, spider arms, spider-ring frame,

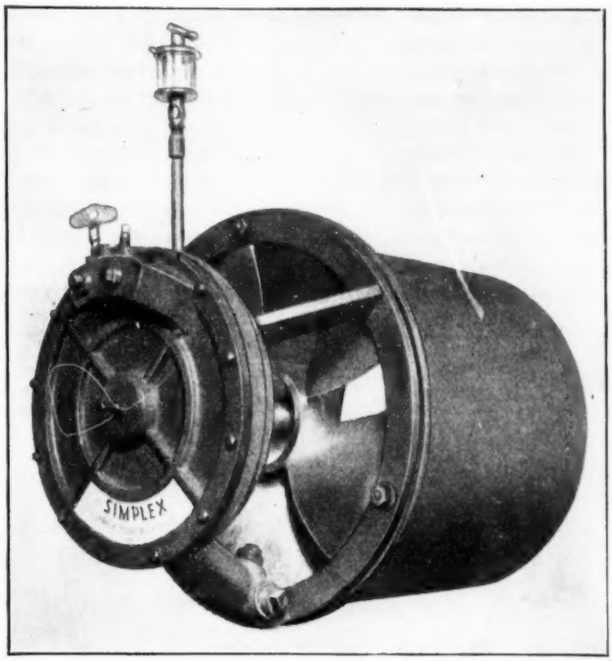


FIG. 1. EXTERIOR OF SIMPLEX BLOWER

exhaust pipe and its connections and port bosses in one casting. The turbine ring, rotor and the rotor hub consist of one bronze casting. Fig. 2 shows the various parts that make up the complete unit.

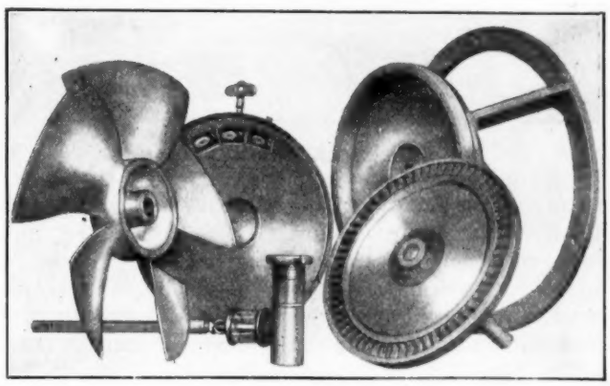


FIG. 2. PARTS OF SIMPLEX BLOWER

Both the bronze rotor and the aluminum propeller are mounted on the same shaft, which revolves in a bushing at the inner end of which the shaft bore is enlarged and fits closely over a collar of the shaft, and a steel washer is

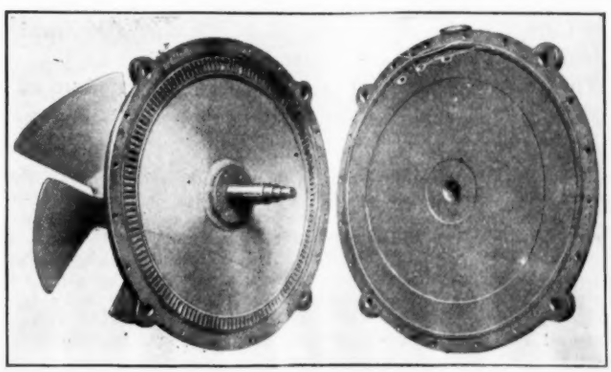


FIG. 3. POWER TURBINE CASING

placed between this collar and the bearing wall formed by the enlargement. This construction takes up the end thrust of the shaft in the direction of the propeiler. The end of the bushing presses against packing that is placed around the shaft.

On the turbine end the steam chest is made small in order to avoid radiation and condensation losses. Individual nozzle control is provided through three separate steam channels, the steam supply being controlled by separate valves. The exhaust passes out of the turbine exhaust chamber through a cored bracket arm that serves

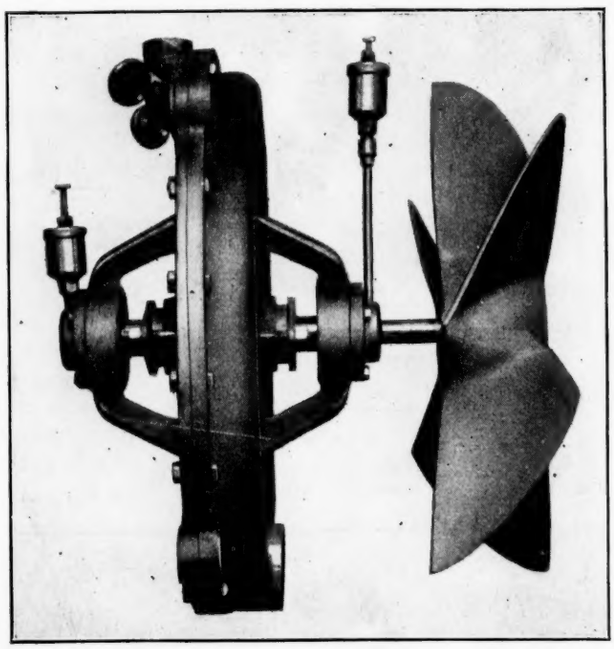


FIG. 4. POWER TURBO-BLOWER

as an exhaust outlet, and the exhaust steam enters witl the air into the furnace ashpit.

In the "Power" turbo-blower, which is designed to supply air sufficient for boilers up to 600-hp. capacity, the shaft is supported by two bearings located on the outboard hubs of the turbine casing and supported by bracket arms.

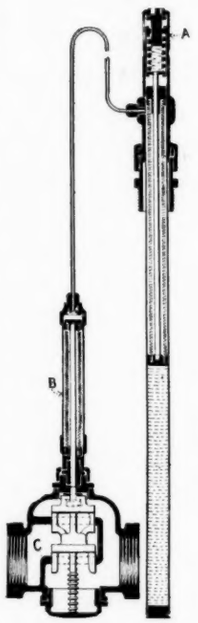
The halves of the turbine casing, Fig. 3, are bolted together. At the top of the casing a small steam chest is provided with three channels leading from the main steam port to the nozzles. These nozzles are independently controlled by needle valves to allow of opening the blower at different speeds without reducing the steam pressure at the nozzle. Exhaust steam leaves the casing at the bottom and goes to the boiler ashpit with the air.

Connections are made between the fan and turbine blower by means of steel distance pieces and studs, Fig. 4. The rotor is bronze, and the propeller is an aluminum alloy.

Spontaneous Combustion is the result of a remarkable tendency observable in tissues and cotton when moistened with oil, to become heated when oxidation sets in, and active combustion often follows when this tendency to take fire is neglected. A wad of cotton used for rubbing a painting has been known to take fire when thrown through the air. The waste from vulcanized rubber, when thrown in a damp condition into a pile, takes fire spontaneously. Masses of coal stored in yards frequently take fire from spontaneous combustion without any spark of fire being applied to the mass. It is good to know such things and to guard against mysterious fires.—"Railway and Locomotive Engineering."

Improved Sarco Temperature Control Regulator

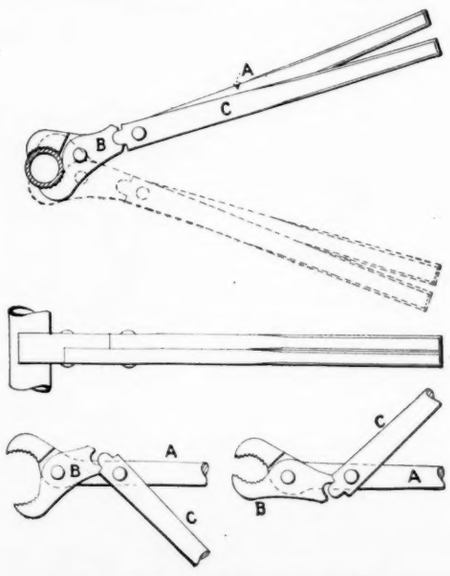
The Sarco Co., Woolworth Building, New York City, has recently placed on the market a modified type of its temperature regulator containing a number of improvements shown in the illustration. The general principle of the regulator is more or less known, as it has been previously described. The apparatus relies on the expansion of a heavy hydrocarbon oil operating a spirally corrugated tube to which a piston is attached to operate a balanced type of valve. Among the new features the regulating head *A* is equipped with an extra-heavy spring between the adjusting screw and the top of the adjusting piston. This spring tightens up and allows for any expansion of the oil in excess of that necessary for closing the valve. This necessity can arise because as much as 2,000 lb. pressure can be generated within the thermostat by exposing it to heat. The spring protects the regulator from being strained should the thermo element be subjected to a higher temperature than that for which it is adjusted. The control element *B* has also been modified, the piston now filling the length of the operating tube and providing for the immediate transmission of expansive power to the valve *C*, which, as now used, is of the piston type and furnished with an extra-heavy vanadium-steel spring operating against the closing pressure.



TEMPERATURE REGULATOR

La Rock Pipe Wrench

The pipe wrench shown in the illustration has been designed for use where the clearance between the pipe and a wall is limited or where the pipes are nested and where it is difficult to manipulate the handles.



DETAILS OF PIPE WRENCH

The wrench, manufactured by the Mechanical Specialties Co., Chicago, Ill., consists of but three parts. One side *A* is made the same as any ordinary pipe wrench. The other side consists of a jaw that is pivoted to the solid side as shown. The inner end of the jaw *B* is made with a recess in which a projection from the handle piece *C* meshes. The handle *C* is pivoted to the piece *A*, as shown. The illustrations show the position of the jaw with the handle *C* in various positions.

Connelly Water-Tube Boiler

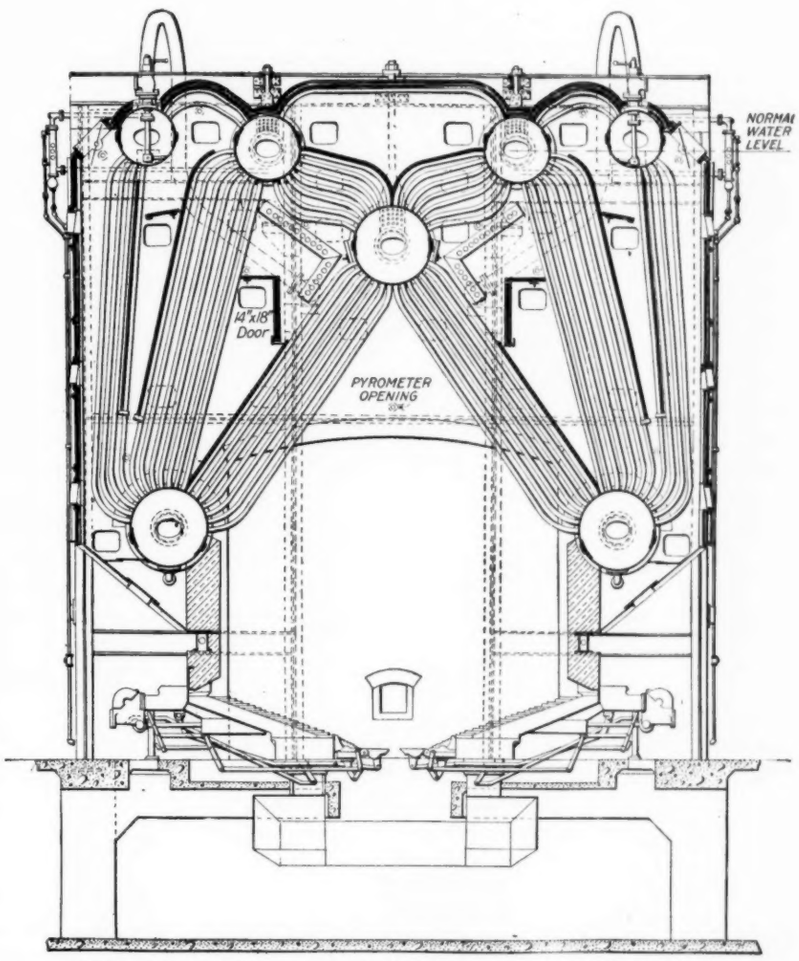
Steam turbines have advanced so rapidly in capacity that to supply them with steam from the ordinary sizes of steam boilers, would require an extremely large boiler room, as regards floor space. Furthermore, small units are not so efficient in operation as large ones and the tendency of present-day practice is toward large boilers for the modern high-pressure power plant. Among others engaged in building large-capacity boilers is the D. Connelly Boiler Co., Cleveland, Ohio. In some respects the Connelly water-tube boiler has characteristics not unlike those of the Stirling. It can be built in units of from 100 to 1,500 hp. in the four-drum type and from 1,000 to 4,000 hp. in the seven-drum design, and is built for a working pressure of 300 lb. In the illustration is shown the 1,000-4,000-hp. design. Remove the two top and one lower right-hand drums and their tube connections to the center drum and the design of the 100-1,500-hp. boiler is shown. In the large unit three of the seven drums are filled with water, the other four being partly filled. The feed water is introduced into the two outside upper drums and flows through the rear bank of tubes into the bottom, or mud drums, where it mixes with the hot water coming down through the middle banks of tubes from the two top inner drums. The water passes from the two mud drums through the inner banks of tubes into the center drum, which is full of water, from which it goes to the two top inner drums, thus completing its cycle. Steam that has been made in the two inner banks of tubes is liberated into the upper inner drums and then passes through the steam-drying tubes into the two outer top drums, from which it is piped to the superheater.

In this boiler the top outer drums to which the steam nozzle outlets are attached are set higher than the inner drums, which gives a greater distance from the water line to the steam outlet nozzle and additional steam space. There is no connection below the water line between the two upper inner drums and the two upper outside drums, therefore there is no commotion of the water in these drums to interfere with the delivery of superheated steam to the generating units.

Owing to the large water capacity and unrestricted circulation, the boiler can be operated under varying loads and at heavy overload capacity, and the water level as indicated by the gage-glass shows no variation when the boiler is operating under fluctuating conditions.

There are but four fixed points in the boiler, the upper drums. The boiler is suspended from a steel frame. All three inside drums are suspended from steel beams and are arranged as to flexibility so as to allow for the expansion of the two inner drums and the tubes. All water tubes are 3¼ in. diameter and are set with alternately wide and narrow spaces, so that any tube in the boiler can be removed without interfering with any other.

Bafflings are made of standard fire tile and no special shapes are required. In the smaller boiler units up to 1,500 hp., the baffling is such that four passes are made by the gases in going from the furnace to the uptake. In the larger sizes the gases make three passes for each half of the boiler. A baffle at the two outside banks of tubes directs the gases to the bottom ends so that they escape at the front of the two outside top drums instead of at the bottom of the last bank of tubes, as is the case with the smaller units.



GENERAL DESIGN OF THE 1,000-4,000-HP. CONNELLY BOILER

Some of the differences in points of construction between the small standard Connelly boiler and the Stirling are that the rear drum of the former is higher than the center and front drums and the tubes connecting the front and center drums are always filled with water, as

is also the front drum, which in the illustration would be the center upper drum. The two highest upper drums of the smaller units are tied together by two sets of drying tubes instead of one as in the Stirling. At normal water level the front drum of the smaller unit is full of water, the center drum is one-half full and the upper rear drum about one-third full. This arrangement, together with that of the baffling, brings the upper portion of the two rear drums and their tie tubes in the path of the hot gases between the third and fourth pass. The result is that the steam in these drums and tubes is somewhat superheated.

The main superheaters of the large unit are placed between the first and second pass, as shown in the illustration. As the water-column connection is to the front drum, which is always full of water, the level showing in the gage-glass does not fluctuate so much as it would if connection was made to the two upper central drums as in the case of the larger units, where the drums are but partly filled with water, which is affected by the circulation from the center drum.

All forged steel manhole plates are hinged to the drumheads. The interior of the boiler setting is accessible through large doors in the front and doors in the side and rear walls, the frames of which are ground and fitted with asbestos gaskets to prevent air leakage.

Soot is blown from the top of the mud drum by providing a cleanout door in the side wall of the boiler setting, where a steam lance can be inserted and the soot blown across into a cast-iron box in the opposite wall. The door of the cleanout box is opened by a lever extending through the rear wall. All pressure parts of the boiler are designed and built to conform with the boiler code of Ohio or that of the American Society of Mechanical Engineers.

Field Coils Carbonized

By E. C. Parham

Other conditions being normal, the amount of current taken by a continuous-current motor armature operating at constant speed depends on the strength of the magnetic field due to the field coils; because on this field strength depends the value of the counter electromotive force, which alone limits the value of the armature current after all the starting resistance has been cut out. Therefore, if for any reason the magnetic field of the motor becomes weak, not only will the armature starting torque be less for a given value of the armature current, but also, when the motor is up to speed, that speed will exceed the speed normally corresponding to the particular load involved and there is likely to be poor commutation owing to the neutral having been shifted as the result of the armature field having become relatively stronger. A weak magnetic field may be due to air-gap irregularities, to shifting of the brushes, to loose polepieces, to reversed field coils, to metallically short-circuited field coils, to grounded field coils (a condition most likely to obtain on a motor that is operated from a ground-return circuit) or to the insulation between the field-coil turns having become carbonized as the result of overload, for example. The field-weakening cause which is to be considered here, is that due to carbonization of the insulation.

An operator sent a motor armature to a repair shop. From the appearance of the commutator it was suspected that the armature had been abused, either as the result of the connected load being excessive or by reason of a weak magnetic field. A man was sent to test the motor fields with a voltmeter and an ammeter. His tests disclosed that with normal line voltage across the field coils, the voltage drops on the coils were equal, showing all the coils to be alike; but the ammeter, which was connected in series with the field coils, showed excessive current. This indicated either that the coils were wound with too large a wire or that their insulation between turns was carbonized. Inspection proved the latter to be the case.

Table To Convert Fahrenheit to Centigrade Readings

By L. Griffin

To convert Fahrenheit to Centigrade: In one of the side sections of the table find first digit or digits of Fahrenheit reading. At the head of the column in which it occurs will be found the hundreds or thousands of the Centigrade value, and in the same horizontal section find the last two

TABLE OF CONVERSION, DEG. C. TO DEG. F. AND VICE VERSA

00	1,000	2,000	3,000	E Deg. C.	0	1	2	3	4	5	6	7	8	9	00	1,000	2,000	3,000	T Deg. C.
				00	32	34	36	37	39	41	43	45	46	48					50
0	18	36	54	01	50	52	54	55	57	59	61	63	64	66	9	27	45	63	51
				02	68	70	72	73	75	77	79	81	82	84					52
				03	86	88	90	91	93	95	97	99							53
				03									00	02					53
1	19	37	55	04	04	06	08	09	11	13	15	17	18	20	10	28	46	64	54
				05	22	24	26	27	29	31	33	35	36	38					55
				06	40	42	44	45	47	49	51	53	54	56					56
				07	58	60	62	63	65	67	69	71	72	74					57
				08	76	78	80	81	83	85	87	89	90	92					58
				09	94	96	98	99											59
				09					01	03	05	07	08	10					59
2	20	38	56	10	12	14	16	17	19	21	23	25	26	28	11	29	47	65	60
				11	30	32	34	35	37	39	41	43	44	46					61
				12	48	50	52	53	55	57	59	61	62	64					62
				13	66	68	70	71	73	75	77	79	80	82					63
				14	84	86	88	89	91	93	95	97	98						64
				14										00					64
3	21	39	57	15	02	04	06	07	09	11	13	15	16	18	12	30	48	66	65
				16	20	22	24	25	27	29	31	33	34	36					66
				17	38	40	42	43	45	47	49	51	52	54					67
				18	56	58	60	61	63	65	67	69	70	72					68
				19	74	76	78	79	81	83	85	87	88	90					69
				20	92	94	96	97	99										70
				20					01	03	05	06	08						70
4	22	40	58	21	10	12	14	15	17	19	21	23	24	26	13	31	49	67	71
				22	28	30	32	33	35	37	39	41	42	44					72
				23	46	48	50	51	53	55	57	59	60	62					73
				24	64	66	68	69	71	73	75	77	78	80					74
				25	82	84	86	87	89	91	93	95	96	98					75
				25	00														75
5	23	41	59	26	00	02	04	05	07	09	11	13	14	16	14	32	50	68	76
				27	18	20	22	23	25	27	29	31	32	34					77
				28	36	38	40	41	43	45	47	49	50	52					78
				29	54	56	58	59	61	63	65	67	68	70					79
				30	72	74	76	77	79	81	83	85	86	88					80
				31	90	92	94	95	97	99									81
				31							01	03	04	06					81
6	24	42	60	32	08	10	12	13	15	17	19	21	22	24	15	33	51	69	82
				33	26	28	30	31	33	35	37	39	40	42					83
				34	44	46	48	49	51	53	55	57	58	60					84
				35	62	64	66	67	69	71	73	75	76	78					85
				36	80	82	84	85	87	89	91	93	94	96					86
				37	98														87
				37		00	02	03	05	07	09	11	12	14					87
7	25	43	61	38	16	18	20	21	23	25	27	29	30	32	16	34	52	70	88
				39	34	36	38	39	41	43	45	47	48	50					89
				40	52	54	56	57	59	61	63	65	66	68					90
				41	70	72	74	75	77	79	81	83	84	86					91
				42	88	90	92	93	95	97	99								92
				42								01	02	04					92
8	26	44	62	43	06	08	10	11	13	15	17	19	20	22	17	35	53	71	93
				44	24	26	28	29	31	33	35	37	38	40					94
				45	42	44	46	47	49	51	53	55	56	58					95
				46	60	62	64	65	67	69	71	73	74	76					96
				47	78	80	82	83	85	87	89	91	92	94					97
				48	96	98													98
				48			00	01	03	05	07	09	10	12					98
9	27	45	63	49	14	16	18	19	21	23	25	27	28	30	18	36	54	72	99
					0	1	2	3	4	5	6	7	8	9					

digits of the Fahrenheit reading, opposite which in column *E* or *T* find the one or two middle digits, and at the head of the column in which the reading is found will be found the last digit (or units) of the Centigrade value.

Example: To convert 4,307 deg. F. to C. Find 43 under 2,000. This is the equivalent thousand, and opposite 07 in the same horizontal section and under *E* find 37, the hundreds and tens, and at the head of the column

(over 07) find 5, the units; or 4,307 deg. F. = 2,375 deg. C.

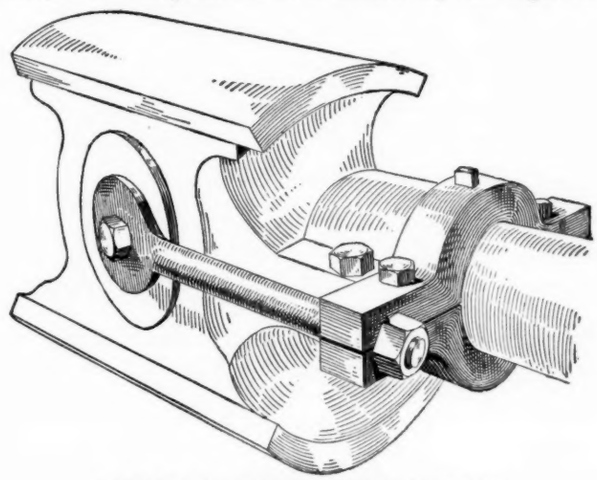
To convert from Centigrade to Fahrenheit, work from the reverse end of the original reading and follow down in the column at the head of which the last digit is found, to a point opposite the next one or two (if there are two) digits in columns *E* or *T*. The value at this point will be the last two digits for the Fahrenheit scale. The value in the same section to the right or left in the hundreds or thousands column will be the hundreds or thousands in deg. F.

Example: To convert 2,375 deg. C to F. Under 5 and opposite 37 find 07, and in the same section at the left under 2,000 find 43. Answer, 4,307 deg. Or if the Centigrade reading is 375, then under 5 and opposite 37 find 07, and in the same section at the left under the hundreds find 7. Answer, 707 deg. F.

Emergency Crosshead Repair

When the dry-vacuum pump breaks down, the chances are that the unit it serves will be out of service until repairs have been made, unless there are means of connecting the unit to another condensing outfit. This is seldom possible where jet or barometric condensers are used, each generating unit being served by a separate condenser.

In one instance the piston rod of a 12&30x18-in. dry-air pump failed by stripping the threads in the crosshead. The pump ran in conjunction with a 3,000-kw. steam turbine that was required to be in service day and night except Sundays. The problem was how to get the air pump in service in the least possible time.



DETAILS OF EMERGENCY CROSSHEAD

In the original design of crosshead the threaded end of the piston rod was secured by clamping it in the slotted end of the crosshead. No jam-nut was used to prevent the rod from working loose.

The delay that getting a new piston would cause was not to be thought of, and John Cooser, chief engineer of the power plant where the accident happened, at the Luzerne mines of the Rochester & Pittsburgh Coal Co., Luzerne, Penn., devised the arrangement shown in the illustration.

As the holding power of the threaded end of the piston could not be depended on, some kind of an auxiliary crosshead had to be designed. This is how it was done: The wristpin was drilled and tapped for a 1½-in. bolt at both ends. Two 1½-in. bolts each 14 in. long were threaded at one end and flattened at the other, making that end 4½ in. wide with the end rounded. A 1½-in. hole was drilled in each. The bolts were threaded far enough to take two nuts and the crosshead clamp.

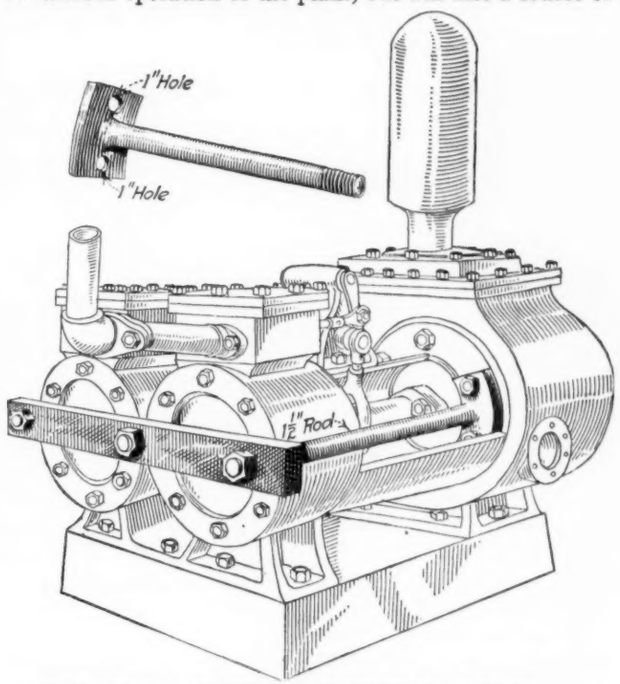
The crosshead clamp was made 3 in. wide of 1-in. thick steel stock and forged at the center to clamp the 2¼-in. rod. Holes were drilled for ¾-in. clamping bolts, and a keyway was cut for keying the piston rod. The rod was also fitted with a keyway.

The job when finished looked like the assembled sketch. From the time the accident happened until the air pump was again in operation was 16 hours. The pump ran with its makeshift crosshead for nearly two months when a new one and piston rod replaced it.

Pump-Frame Reinforcement

Trouble was experienced with the breaking of the frame distance piece between the cylinders of two duplex boiler-feed pumps. This was evidently caused by insufficient material, and the engineer took that view of the case.

This occasional breakage was not only a menace to the continuous operation of the plant, but was also a source of worry to all concerned. As a precaution against a recurrence of the trouble, the engineer had three 1½-in. rods made with a tie head forged on one end and the other end threaded. The tie heads were drilled for 1-in. capscrews. A bar of forged steel was made 2¼ in. thick and long enough to reach across the steam-cylinder heads of the pump, allowing for a 1½-in. hole at each end and one for the center rod.



PUMP FRAME REINFORCED WITH STAY RODS

The holes in the tie-head end of the brace were spaced so that the capscrews would align with the stud-bolt holes in the distance piece. When the job was completed, the pump was stayed as shown in the illustration, the stay rods being bolted at one end and passing through the forged cross-piece at the other. When the nuts were tightened, the distance pieces were relieved of a part of the strain to which they were subjected. No trouble has since been experienced from breaking distance pieces.

JUST FOR FUN

Figures Never Lie

[The following comes to us from a source unknown.—Editor.]

The engineer approached the Boss on the matter of a "raise," claiming he felt entitled to it, having worked hard all the preceding years.

With an expression of surprise, the Boss turned and faced him; then, without answering, he reached for a pencil and with an air of confidence in his man's good sense and honesty, he began, "You know the year has 365 days, don't you?"

Engineer: (Slowly) "Yes."

Boss: "You sleep 8 hours a day; that makes 121 days a year."

E. "Yes."

B. "Taking 121 off 365 leaves 244 days, you understand?"

E. "Yes."

B. "You have 8 hours free time each day, or 121 days a year; taking that off 244 days leaves 123 days."

E. "Yes."

B. "Now you have 52 Sundays off; that leaves you 71 days."

E. "Ye-es."

B. "And 52 Saturdays of half a day each, or 26 days from 71 leaves 45 days."

E. "Ye-e-es."

B. "You take 1 hour or more for lunch daily, making 30 days a year, or leaving you 15 out of the 45 days."

E. "Ye-e-e-es."

B. "You get 2 weeks vacation every year; taking those off the 15 days leaves 1 day; on Fourth of July you don't work. *Now* can you tell me why you expect a raise?"

E. (Dazedly) "I haven't been working at all."

B. *"Certainly not!"*

A Deep Mine Shaft

Three Western engineers were discussing deep mine shafts, and two of them had told in glowing terms about mines in which they had worked.

"You fellows never saw a mine shaft," said the third. "Jump into my car and I'll take you to a regular mine."

They were soon at the mouth of the shaft. The hoisting engine was revolving at a tremendously high rate of speed, and to their horror they saw that the engineman was sound asleep in his chair. Both men rushed to the sleeping man, crying, "Wake up, man! You'll pull the cage through the roof!"

"What day is this?" the engineman asked sleepily.

"It's Tuesday, but stop her, quick!"

"Aw g'wan," he replied, disgustedly, settling himself back in his chair. "She won't be up till Friday."—Harlan A. Russell, Lawrence, Kan.

Cylinder Oil Containing Acid will probably cause pitting of the cylinder walls, hence it is advisable to watch the condition of cylinders after changing from an oil that is known to give good satisfaction to another grade.

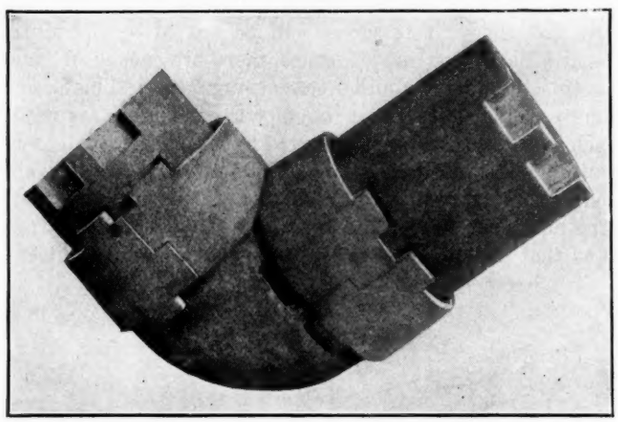
"J-M Aertite" Coating for Boiler Walls

Engineers will be interested in "J-M Aertite," a new boiler-setting coating placed on the market by the H. W. Johns-Manville Co., Madison Ave. and 41st St., New York City.

The coating is applied to the outside of the boiler setting to eliminate air infiltration. It provides a tough, rubbery coating that remains tight on account of its adhesive and ductile qualities. It is applied by troweling, and the best results are obtained by keeping it as near $\frac{1}{16}$ in. thick as possible.

Blowoff Pipe Covering

One of the boiler fittings that is exposed to the intense heat of the furnace is the blowoff-pipe connection. Many engineers advocate protecting it with some sort of covering. For this purpose the Interlocking Pipe Covering Co., 320 East 29th St., New York City, manufactures a sectional covering such as is shown in the illustration. The covering is molded to fit elbows and in sleeves for the pipe lengths. Elbow coverings are made in 2, 2½ and 3 in. sizes. A covering for the bolt and bridge of the rear-end handhole plate is also made to protect them from the intense heat of the furnace.



BLOWOFF PIPE COVERING

One Formula for the Flow of Steam in Pipes is based upon pipes under 100 ft. long and between 4 and 8 in. diameter and an allowable steam velocity of about 6,000 ft. per min. (V_1), friction and other causes of drop in pressure disregarded. Thus,

$$d = 0.175 \left(\frac{w}{y}\right)^{\frac{1}{2}}$$

in which

d = Diameter of pipe, in.;
y = Density of steam, lb. per cu.ft.;
w = Weight of steam flowing in lb. per min.;
V_1 = Velocity 6,000 and V_2 any assumed velocity;
0.175 = Constant (C_1) given and C_2 new constant.

For higher or lower velocity than 6,000 ft. per min., the constant in the equation is changed accordingly by simple proportion for $V_1 \times C_1 \div V_2 = C_2$, or for, say, 3,500 ft. per min., is $6{,}000 \times 0.175 \div 3{,}500 = 0.3$ as the constant for that velocity. The constant for any other velocity is similarly found, or the velocity by any constant is found by $V_1 \times C_1 \div C_2 = V_2$. Example: $6{,}000 \times 0.175 \div 0.3 = 3{,}500$. The constant for the present-day higher velocity of, say, 15,000 ft. per min. is therefore 0.07 by the foregoing process. In large diameters the velocity may be increased with advantage, but for small diameters or great lengths, friction and condensation cause considerable drop in pressure, and some one of the more elaborate approved formulas may be used.

Editorials

California Engine-Safety Orders

The Industrial Accident Commission of California has issued orders relative to safety provisions in engine rooms. These went into effect August first and appear elsewhere in this issue.

These orders are all commendable, but one of the best of the lot is that requiring permanent ladders where stairways are needed but are not practicable. Specifically this order does not apply to boiler rooms, but Mr. Brownell, superintendent of safety for the commission, in response to our inquiry stated that the commission's inspectors required permanent ladders to be attached to each boiler setting.

This is indeed a most commendable requirement. It is idle to belittle the potential danger of boilers and their steam mains under pressure. The conditions make it imperative that every facility be provided for escape of the worker who may be on top of them. But in how many plants does one find ladders arranged to make it possible to get quickly down from or up to the tops of boilers? Unfortunately all too many boiler rooms are without a ladder of any kind, while others have ladders which are supposed never to be removed from the room, but which require to be hunted when wanted.

Plant designers should always provide permanent ladders on boiler settings. Those cities and states that have safety laws should follow California's lead and require metal ladders for these places.

Ventilating Electric Machines

Elsewhere in this issue is an article dealing with the ventilation of electrical machinery. The development of any art is usually interesting, and the art of ventilating electrical machines is no exception. The early types seem to have been designed and constructed to prevent the heat from getting out of the different parts rather than to liberate it. In those early days the question of prime importance to the designer and the manufacturer was to get a machine that would make a fairly good showing for continuous operation rather than maximum horsepower and efficiency for a given weight of material. Consequently, the ends of and sometimes the entire armature windings were covered with canvas and this thoroughly saturated with shellac or other insulating compound to protect the armature from injury. However, a radical change has taken place in the design of electrical machinery, and today many small-sized machines are equipped with ventilating ducts in the armature core and have a fan mounted directly on the armature shaft within the machine for air circulation through the various parts.

The problem of ventilating slow-speed generators directly connected to reciprocating engines was not a serious one. In this case it was more a matter of efficiency than of temperature rise. With the advent of the high-speed turbine-driven machines the capacities increased many times for a given weight and the problem of ventilation became one of vital importance, as it was no longer a question of efficiency, but of liberating the heat generated in the machine. Turbo-alternators represent the extreme in high-speed machines, and proper ventilation is one of the most difficult features in their design. Therefore, as Mr. Fox points out, it has been necessary to carry forced cooling to its most complete application in this type of apparatus. The effectiveness of such ventilation is appreciated when it is realized that the modern steam-turbine-driven alternator without some special means of ventilation would reach a dangerous temperature, if run without any load whatever at normal or even less than normal voltage.

Canada and Niagara Power

Attention is again drawn to Niagara power by the recent action of the Canadian government in compelling the Canadian Niagara Power Co., through threats of prohibiting the exportation of power, to supply the Ontario Hydro-Electric Commission with the energy it desired. The latter, it will be recalled, is a public undertaking which has done much to extend the use of electricity at low cost throughout the province of Ontario.

It appears that the commission had contracted to supply a number of municipalities and, being unable to secure the power from the hydro-electric company at a figure considerably less than the latter was receiving for the energy exported to the American side, appealed to the Dominion government, with the result noted. The incident emphasizes the attitude of the Canadian government as a reflection of public opinion—the municipalities needed the electricity and were able to force their demands upon the power company, although it meant a monetary sacrifice to the latter. It is coincidental that this, an example of how such problems are dealt with in Canada, should have occurred while our Senate and House are endeavoring, through a conference committee, to reach an agreement on a national water-power policy.

There is another and equally important lesson to be learned from the action of the Canadian authorities. Although the power companies on the Canadian side of the Falls are owned largely by American interests and about one-half their output is exported to the American side as allowed by the treaty of 1910, there is a growing demand for power in Canada backed by public sentiment that more industries should be attracted to that side of the border. Moreover, the recent action of the Ottawa authorities indicates that they are prepared to back up the "Canada first" idea. This is only natural; indeed they would be remiss in their public duty if they did not. Nevertheless, now that American industries have thus been forewarned steps should be taken to meet such exigencies.

The total diversion of water at Niagara as allowed by the treaty between Great Britain and the United States is fifty-six thousand cubic feet per second, thirty-six thousand cubic feet on the Canadian side and twenty thousand on the American side. However, the War Department will permit a diversion of only fifteen thousand, six hundred cubic feet per second, as formerly allowed under the Burton act. The difference, four thousand, four hundred cubic feet, is equivalent to about eighty thousand horsepower, and it is understood that the power companies on the American side have equipment installed capable of utilizing this surplus water. While this additional power would not be large compared with the demand, still it would help out until some means is adopted for extracting more power without destroying the scenic beauty of the Falls. The action of the War Department in withholding the use of the surplus water can be construed only as lack of confidence in the engineers who made the survey for the diversion allowed by the international treaty.

Feed Water at the Mines

When a commercial power plant is built, it may or it may not be close to the center of distribution. It must, however, be at a point where an abundance of suitable water can be obtained for boiler feed and for condensing purposes.

A mine power plant is a different proposition. Unless it is an electrically operated property, the power house and especially the boilers must be close to the mine shaft so as to pipe steam conveniently to the fans, the hoisting apparatus and the steam pumps in the mine. The availability of a supply of water is one of the last things to be considered, although the mine operation is dependent upon the boiler plant.

In some few instances a sufficient supply of a good quality of water is available for boiler and other power-plant uses. Generally, however, that at hand is either unfit in the first place or has been made so by contamination with mine water. In addition to the ordinary impurities common to the average water supply there are sulphuric acid and sulphate of iron, lime and magnesia. As is well known, these last two sulphates produce a hard scale in a boiler, especially the sulphate of lime.

Sulphuric acid that is set free, as well as that introduced in a free state with the feed water, will attack the metal of the boiler. The acid liberated by the heat from the feed water repeats its attack on the metal, and the amount of acid in the boiler is constantly being increased by the addition of new feed water. There are sometimes other corrosive acids, but the most destructive to the boilers, heaters and piping is sulphuric acid.

In power plants using such contaminated feed water the engineer has not only to combat the evils that arise from the formation of scale in the boilers, but he must also fight the attack of acids.

Many makeshift attempts are made to purify water from the vicinity of the mines, but with few exceptions the results are not satisfactory. One reason is that the character of the water changes from day to day, and the feeding of a certain amount of chemicals one day for one composition of water will not be proportioned for that to be treated a few days later.

Experience has demonstrated that the water treated each day should be purified with chemicals that will take care of that particular water; the amount of chemicals to be used the day following may be more or it may be less. Administering the proper proportion to the various qualities of water will so purify it as to make it most suitable for boiler feed under the existing conditions.

Although the losses occasioned by the presence of heavy scale in mine steam boilers have been given but scant attention on the part of coal-mine operators, the excessive coal consumption over what would be secured with properly set boilers, free from scale and the corrosive action of acids in the feed water, would be a big surprise if the difference expressed in dollars and cents were really known.

It is a simple matter to receive a report that number four boiler has been laid off for retubing on account of pitting, but tubes cost money and in the majority of coal-mine power plants it would doubtless be a money-saving proposition if an efficient water-softening plant were put in operation.

"Saving at the spigot and wasting at the bung" is what the plant owner does who puts all his endeavor into reducing steam consumption without striving to increase the efficiency of his steam production. Under the average conditions as they exist today it is as easy to save twenty per cent. in the boiler room as two per cent. in the engine room. Yet it is the common thing to find a clean, economical installation of engines or turbines supplied by a dirty, wasteful steam-generating plant. This is just as poor taste, to say the least, as to put dress clothes on over soiled or torn underwear, but worse than that, it is positively inexcusable extravagance to harbor a coal-wasting steam plant.

One of our readers is building a five-thousand dollar coal bunker under the sidewalk chiefly to avoid being compelled to buy coal that is twenty per cent. ice, as he claims it is for days after heavy snowstorms—which is one way of "freezing out" the coal man.

One of the best power-plant axioms we've heard in a long time is this: "Find and produce those conditions most beneficial, economically and otherwise, to the plant, then provide so that these conditions will be normal and easiest for the men to maintain."

The next council of the American Society of Mechanical Engineers will be largely professorial in character. Of the eight names presented by the nominating committee, five, including the president, are professors.

As an indication of the progress of mechanical refrigeration, N. A. Borodine, in a bulletin of the Russian Committee on Refrigeration, proposes studies in that art for students in commercial high schools.

Enthusiasm, energy and persistence often achieve more than ability. Steam engineering is no exception among the arts to which this applies.

Correspondence

Filtration of Softened Water

I have just read with interest Mr. Vater's article on "Filtration of Softened Water," which appeared in *Power* for May 9, 1916, and I am somewhat surprised at some of the statements. The following discussion is sent in the hope of correcting some of the impressions that may have been formed after reading the article.

So much stress is laid on the savings to be effected by a suitable filter to arrest and separate matter mechanically suspended in water that the user is likely to be misled and look upon the installation of a filter as a cure-all for sludge troubles. Such is not always the case, and more often the remedy should be applied to the operation of the softener or the installation of an adequate softening apparatus. There are many plants having so-called purifiers where the passage of water through them is so rapid that the chemical reactions are not complete before the water is sent to the filter beds, and not infrequently the after reactions seriously clog the filters or take place after the water has reached the hotwell or boilers. The trouble is not with the filter, but is either caused by the user trying to get along with a plant too small for the requirements, or by inherent defects in the softening apparatus.

The proper function of a filter in connection with a water-softening apparatus is to remove from the treated water only the last traces of sludge and suspended matter. Indeed, if the water were correctly treated and sufficient time allowed for the completion of the chemical reactions and subsequent clarification by precipitations, the filter, in many plants, might be done away with without filling the boilers, feed lines, regulators, flow meters, etc., with deposit, provided the water were not drawn from the bottom of the tank where it might be contaminated by the sediment. Deposit such as that Mr. Vater mentions as occurring in boilers, feed lines, regulators and flow meters is usual with an inadequate reaction. The water should be practically clear before it reaches the filter, and a filter is only as efficient as the character of the applied water. When, therefore, it is called upon to perform the kind of duty suggested by Mr. Vater's article, the user might better look into the operation of his plant or consider the cost of an adequate softening system, rather than install a filter in the vain hope of eliminating troubles by trying to make a purifier out of a sand bed.

As a purifier a filter is a failure. If the water has not been properly treated and sufficient time allowed for the reactions to be completed and for sedimentation to take place before going to the hotwell, no amount of interposed filters will prevent the formation of sludge in the boilers because no filter can remove matter that is not in suspension. In some cases it cannot altogether remove matter that is in suspension, and it then becomes necessary to prepare the water for effective filtration. Sometimes, with a fully treated water, so far as lime and soda-ash are concerned, it is necessary, even in plants in which the water has had a full static period for precipitation, to add iron sulphate to the water to properly coagulate it for filtration.

When iron sulphate is used correctly for coagulation in an alkaline water—and no chemicals should ever be applied without proper tests or supervision—it cannot in any way injure the boilers.

Mr. Vater's statement, "that sulphate of iron does not coagulate until it has released its sulphuric acid, becoming iron hydrate and leaving free sulphuric acid in the water," to be exact should be stated thus: By the addition of iron sulphate to an alkaline water, it results in the formation of iron hydrate and the sulphates of lime or soda. At no stage of the coagulating process is free sulphuric acid released. That sulphuric acid exists in the free state in an alkaline water, which always obtains where correct amounts of lime and soda ash have been applied, is contrary to all the laws of chemistry, and so far as the writer is aware, is an impossibility. The iron sulphate reacts directly with the calcium and sodium salts, forming sulphates which are extremely stable compounds and do not dissociate even under the conditions of temperature and pressure in the boilers.

The inference drawn from the "serious depreciation" in the plant referred to is that free sulphuric acid caused the trouble; but it is extremely doubtful if such was the case, unless iron sulphate was used carelessly and without proper knowledge of its chemical nature. The difficulty was more than likely caused by the poor operation of the plant, or an inefficient softening system, in which case the addition of a filter would not necessarily have solved the problem. W. A. CRONKITE.

Appleton, Wis.

Viscosity in Cylinder Oils

Mr. Carpenter's article in the issue of Apr. 11 is interesting, especially the last paragraph: "Viscosity is the great humbug test featured in all specifications," etc.

Why he has picked out the viscosity test to be condemned I am at a loss to know. Whether specifications are worth anything or not depends upon who writes them. The viscosity test is only one of many that make up the specification, and alone it means little or nothing. One might as well state the amount of carbon only in writing the specification for iron or steel, leaving out sulphur, phosphorus, silica, etc. This procedure is not followed, nor is the viscosity test of a cylinder oil featured to a greater extent than all the other tests. When all the tests of a cylinder oil, or any lubricating oil, are taken into consideration together, they serve as a guide to a competent man. To leave out the viscosity test would make the specification inadequate.

The viscosity test is so much thought of by many oil refiners that it is made use of daily. These refiners know that if the oil is made in a certain manner and has the viscosity they wish it to have (other physical tests not being neglected), it will meet the requirements of the engineer.

When the mineral lubricating-oil business started forty years ago or more, there were no specfications or viscosity tests. Refiners endeavored to manufacture a first-class cylinder oil. After several attempts this was accomplished, and an oil acceptable to the engineer was placed on the market. It then became necessary to duplicate orders. How was this to be done? The refiner could not depend on the stillman altogether, nor could each grade, or batch, be tried out on an engine or machine before being sold. The physical tests as we know them today were gradually adopted, the viscosity test being only one.

While the methods at present in use for testing oils—other than testing them in service—could be improved, it must be said that they have served their purpose well and are still being made use of. In the hands of a man who knows his business the usual tests tell him how the oil was made, where it can be used to the best advantage, and sometimes the crude oil it came from. The engineer (the consumer) finally gains the benefit of all this testing, though indirectly perhaps. The viscosity test is not such a "humbug" after all.

To continue and discuss Mr. Carpenter's reply in the issue of July 18: After describing the Saybolt viscosimeter and the manner of using it, he says "there is nothing in this physical test that reproduces the physical conditions under which the cylinder oil is to lubricate."

I do not think that a claim has been put forward that this test reproduces the physical conditions mentioned. It simply happens to be one of the tests that are taken into consideration along with several others when the quality of a cylinder oil is under discussion.

Mr. Carpenter knows that it does not reproduce the physical conditions under which a cylinder oil is used, yet I shall be much surprised if he does not make use of this test daily in his laboratory, and should he discontinue its use, I am afraid he would be greatly at a loss.

The determination of viscosities at temperatures above 212 deg. F. is not new. Messrs. Wells and Taggart in 1903 examined 400 oils. The report on 100 of them is published in their book "Cylinder Oil and Cylinder Lubrication," published by the Henry Wells Oil Co., Manchester, England. The oils they examined were tested in a viscosimeter at temperatures from 212 deg. F. to 600 deg. F. These investigators noted a quick drop in the viscosity of all cylinder oils above 212 deg. F.

Tallow is discussed by Mr. Carpenter. There is no connection between a discussion on viscosity and a discussion on the use of tallow. Tallow, tallow oil or other high-grade animal oils will continue to be used in small amounts with mineral cylinder oils, whether or not they lose their viscosity above 212 deg. F. to a greater extent than the mineral oils.

These animal oils are added to the mineral cylinder oils to give to the latter a property much of which they do not possess, one that is required for cylinder lubrication and which for the want of a better name I must call "greasiness." It is a fact that these animal oils help the mineral oils to cling to the surface of the wet cylinder more readily than the mineral oil will do alone.

We still have much to learn about lubricating oils.

One of the most acceptable additions to our knowledge on cylinder oils and cylinder lubrication appears in this same issue of *Power,* namely, July 18. I refer to the article on "The Fractional Distillation of Lubricating Oils," by J. G. O'Neill. This I consider one of the best that have appeared on the subject.

The method of examination described by Mr. O'Neill comes closer to working conditions than any other. A careful study of this article will show the value of the viscosity test—as well as some of the other tests—when made use of properly and the results interpreted correctly. The viscosity test in this case becomes a valuable aid.

I do not agree with Mr. Carpenter's statement that "If viscosity is viscosity, what difference does it make how the oil gets it?" The viscosity of any lubricating oil should be honestly acquired and not obtained by adulterating with oleate of aluminum or any other oil thickener to give a false viscosity. Artificially thickened oils have their use—maybe—and are so labeled, but no oil merchant who values his reputation would take a spindle oil, thicken it with oleate of aluminum, discolor it a little and sell it for cylinder oil. Yet by this method he can obtain a viscosity equal to that of any cylinder oil he wishes. The ruse, however, would soon be discovered.

I doubt Mr. Carpenter's statement that artificially thickened cylinder oils can be made and not be detected in any laboratory. The matter is not worth discussing; there are too many grades on the market now that are honestly made.

J. Motion,
Jersey City, N. J. Petroleum Chemist.

Referring to Charles E. Carpenter's letter on the viscosity of cylinder oils, in the July 18 issue, I beg to say that Mr. Carpenter's denials do not alter the facts. My statement as to claims made by persons interested in exploiting cylinder oils containing animal oils was not aimed at Mr. Carpenter; it is a statement of facts, based on my own experience.

Mr. Carpenter's query, "If viscosity is viscosity, what difference does it make how the oil gets it?" makes discussion of the subject futile.

As to filtered oil, I cheerfully vindicate Mr. Carpenter's supposition and deny that filtration improves the lubricating quality of all cylinder oils. It improves the quality of those oils which contain substances devoid of lubricating value; the number is large, but there are some exceptions.

Mr. Carpenter is welcome to his opinion regarding the specific gravity of cylinder oils; my statement is based on facts derived from tests made with steam engines, which a library-full of denials cannot change.

Cecil P. Poole,
Atlanta, Ga. City Mechanical Engineer.

I have been following with interest the discussion on cylinder lubrication by Messrs. Carpenter, Fairbanks and Poole. I have found by experience that Mr. Fairbanks is right in saying that good lubrication depends more on the manner of applying oil to the surfaces than on the viscosity or flash point of the oil.

Mr. Carpenter is right in saying that the only satisfactory test for cylinder oil is to try it out in the engine for a week or longer, then the lubricant that is most suitable shows the best coating of oil on the working surfaces, especially on the bottom and top of the cylinder and on the face of the valve seats.

I am using reclaimed cylinder oil, adding about 25 per cent. of new oil to the reclaimed oil. The flash-point of the original oil is 520 deg. F. and the viscosity is very high, while the reclaimed oil as used has a flash point of about 485 to 490 deg. F. and a low viscosity, yet the reclaimed oil shows better lubrication with less carbon deposit than the original oil. The temperature of the steam is about 475 deg. F. at the throttle, therefore the viscosity of the oil must be very low at the temperature of use.

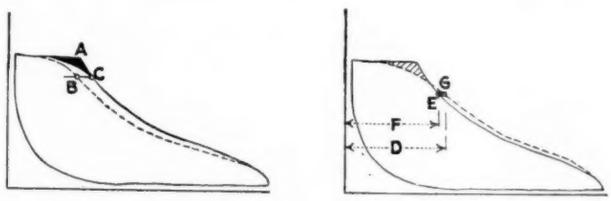
I do not agree with Mr. Carpenter that it makes no difference how oil gets its viscosity, because the material used to give oil viscosity may be detrimental to good lubrication. I think we are all agreed on one point, however; namely, that proper methods of introduction are needed to obtain good lubrication and that the use of oil under working conditions is the most satisfactory test.

A. A. BLANCHARD.

Ringwood Manor, N. J.

Effect of Early Cutoff on Wiredrawing

I read the interesting article by Prof. T. T. Eyre on wiredrawing, in the July 4 issue, page 11, and find that his conclusions are based on the common error of calling the reduction in the power a loss. Two indicator diagrams showing different mean effective pressures cannot



FIGS. 1 AND 2. SHOWING RESULT OF WIREDRAWING

be compared, unless the steam consumption is considered likewise. By wiredrawing two effects are produced: One is that the pressure at the point of cutoff is reduced and the steam consumption diminished; the other is that the mean effective pressure is reduced and the power is diminished. The effect on the economy will therefore be slight. In fact, in his Figs. 12 and 13 the m.e.p. appears to be equal to or greater than that in the theoretical diagram, while the pressure at the point of cutoff is considerably reduced, indicating a gain in economy by throttling.

The true effect of the throttling is illustrated by Figs. 1 and 2. In Fig. 1 the assumed point of cutoff is at *A*. Wiredrawing would reduce the pressure at the point of cutoff from *A* to *B*, but as this reduces the power, the point of cutoff must shift to *C*, at which point the same amount of steam is present in the cylinder as would be with the governor cutting off at *A*. But owing to the later cutoff, the power indicated by the shaded triangle is lost and therefore must be made up by a still later cutoff, say at point *G*, Fig. 2, this point being selected so as to equalize the loss and gain in power. If a horizontal line be drawn through *G*, the theoretical expansion line is struck at *E*, and the distances *F* and *D* of these two points from the zero line furnish a measure for the increase in steam consumption, which increase is the loss due to throttling. It is obvious that this loss is only a trifle; in fact, it may be compensated for by the superheat produced by the throttling effect. The real reason for the higher economy of the Corliss engine is found in the shorter ports and other improvements in the design, rather than this slight gain in steam consumption due to the eliminated throttling effect.

F. F. NICKEL.

Harrison, N. J.

Developing Efficiency in Central Stations

In the article, "Developing Efficiency in Central Stations," page 8 of the July 4 issue of *Power*, the general argument of the text is all right, although it should be read with judgment, but some of the figures given are misleading and some of them are wrong.

For instance, Fig. 1 purports to show the B.t.u. loss with different flue temperatures and percentages of CO_2, but this can be true for only one weight of air per pound of fuel, which, judging from the rest of the text, has been taken as 12 lb.—that is, assuming the fuel pure carbon. But if such were the case, the heat developed would be 14,545 B.t.u. per pound and the percentage loss would be 20.6 per cent. instead of 25 per cent. Or, if the fuel developed but 12,000 B.t.u. per pound owing to ash and moisture content, the theoretical air supply would have been about 9.9 lb. of air per pound of fuel and the percentage loss still approximately 20.6 per cent.

In Fig. 2 the words "Good Practice" and "Average Practice" are misleading, for it is not good practice to try to run with less than the theoretical air supply for complete combustion. My experience has been that even with boilers having extremely large combustion space (lower tube 6 to 8 ft. above the grates) it is difficult to run with less than 30 per cent. air supply above that required for complete combustion without getting enough unburned carbon monoxide and other gases to cause serious loss. Where boilers are set low or the grates or stokers are such that it is difficult to carry a uniform fire, it may be more economical to run with an average of 60 to 75 per cent. excess air rather than incur the losses due to incomplete combustion with a less excess of air and higher CO_2. The temperatures of combustion are based on the assumption that the specific heat of the products of combustion is constant, whereas it increases at the higher temperatures so that the maximum theoretical temperature with complete combustion without excess air is 3,700 instead of 4,800 deg. F.

Fig. 4 is in error, as may be seen by referring to Table 3 on page 31 of the same issue, as for all grades of oil at 15 per cent. CO_2 there is an extremely small percentage (1 to 6 per cent.) of air above that theoretically necessary for combustion whereas Fig. 4 shows this to be about 45 per cent.[1]

The figures given on the right side of the curve, "Per cent. Preventable Loss," can be true for only one flue temperature, and this is subject to great variation, depending not only on the cleanliness of the tubes, but on the amount of excess air. The excess air passes through the furnace and is brought up to furnace temperature, after which it is difficult for the boiler to absorb all the heat. The temperature is low if cold air is leaking in through the setting and merely diluting the gases from which part of the heat has already been absorbed.

[1]The wording at the left of Fig. 4 should read "Per Cent. Total Air," not "Per Cent. Excess Air."

A very useful criterion in data of this kind is that the percentage loss of heat in flue gases, over the ordinary range of boiler flue temperatures, is 2.06 per cent. of the heat of the fuel for each 100 deg. F. temperature of the flue gas above the boiler room for the products of combustion; that is, that part of the flue gases comprising the air theoretically required to burn the fuel, and 1.70 per cent. for 100 per cent. excess air per 100 deg. temperature. H. D. FISHER.

Newton Highlands, Mass.

Heat in Exhaust Steam

Reference was made in an editorial in the issue of June 20 to a quotation by Prof. R. S. Hawley from a statement by the writer regarding the heating value of exhaust steam, which, taken by itself, gives a somewhat wrong impression.

The last part of this quotation reads, "The remaining 20 per cent. is partly converted into work and partly lost through 'cylinder condensation.'" To this should have been added "due to radiation losses."

As stated in the editorial, the water (exclusive of original entrainment) that is discharged in the exhaust steam is due entirely to condensation resulting from the extraction of heat in the performance of work and to radiation and conduction losses from the cylinder.

In speaking of this loss in connection with the value of exhaust steam for heating purposes the writer has sometimes used the term "cylinder condensation," qualifying it with a proper explanation as to its exact meaning. Possibly this was not made sufficiently clear in the article from which the quotation was made, or was simply omitted in using the reference.

The writer wishes to state that his understanding of the matter is in entire agreement with that of the editor.

Jamaica Plain, Mass. C. L. HUBBARD.

Steam from Hot Water

A young engineer friend who is struggling toward a higher-grade certificate often comes to me with problems that he meets with, and because his difficulties are probably the same as those of others of his class, they may interest others.

He is firing a locomotive-type boiler with shavings and has found that after the fire is let out completely the engine will run longer than the steam in the boiler would seem to warrant. He figured the cubic contents of the engine cylinder up to the point of cutoff for both strokes and multiplied by the r.p.m., and found that the volume of the steam space in the boiler would not supply the steam for the length of time the engine continued to run, even by allowing for the expansion of the steam.

Instead of giving him a direct answer, I thought it would be better to put him in the way of discovering the reason for himself, so I told him to try the experiment a couple of times under different conditions: First, to fill the boiler to the top of the glass and have steam at 100 lb., let the fire out and time the run of the engine while the gage pressure was lowered to 20 lb.; that, I explained, would give a small steam space to draw from. Then for the second experiment I told him to have the glass just empty and steam at the same pressure when the fire was let out, and again to time the run of the engine while the pressure was lowered to 20 lb. as before, but with this larger steam space to draw from. I cautioned him not to add any feed water during either test and to have the dampers closed in both cases. "And," I added, "if the water goes out of sight on the second test, it can do no harm, for there is no fire and no hot brickwork; and run the shafting empty for the two evenings to assure the same load in each test."

My experiment on him did not prove as successful as I had hoped, for instead of returning with the answer discovered, he came back mystified more than ever. "Say," he said with a half-laugh, "that darn thing's got me"; then, quite seriously, "Would you believe it, she ran *longest* on the *least* steam—that is, when the water was high." He hadn't caught on to the "why" of it, but the way was cleared for getting the reason so fastened in his mind that it would stay.

"Then, Pete, you have proved this fact—that with engine load and boiler pressure the same in both cases, you had to take more steam out of the small steam space than the big one to lower the steam pressure to the same amount (20 lb.) in each trial."

"Yes."

"You got the *most* steam from the *smallest* steam space at equal pressures," I persisted.

"Y-yes."

"Well, where did that *extra* supply of steam come from?"

"I don't know."

"You *do* know. It didn't come from the boiler shell."

"It must have come out of the water then—it must have," he concluded slowly.

"That's exactly where it did come from, Pete. Didn't the water level fall as the engine ran?"

"Yes, it did—some; but I was so busy with my watch and the steam gage that I didn't look close; but, Chief, you don't mean that water can be evaporated without heat—and once the fire is out the boiler isn't any hotter than the water in it."

"That's perfectly true," I agreed, "but what of the heat already in the water?" Pete looked thoughtful for a moment, and while he hesitated, I added: "That's the key to your problem, Pete. Can't you see now why I had you make that test? You got the *most* steam when you had the *most* water."

"It looks as if you're right, Chief, but it isn't clear. If it's the heat in the water that turns it into steam for the engine, then when the engine is shut down at night and the fire is out, why doesn't the steam rise? It doesn't—I know that."

"Good boy, Pete! Now what is the boiling point of water?"

"It is 212 deg. F."

"And if you add heat beyond that point, what happens?"

"Why, it won't get any hotter, but it will change into steam instead."

"Then you can't make water any hotter than the boiling point?"

"No, Chief, you can't, but the boiling point changes when there's pressure on the water. The more pressure the higher the boiling point goes."

"When you had 100 lb. pressure on it, what would the boiling point be?"

"About 300 deg. F."

"Now, Pete, when you lower the steam pres——"

"Oh, wait—excuse me, I see it now. Just let me *try*. See if I have it."

"Sure thing. I've been expecting you to 'tumble' all along—go to it."

"Why, when the engine lowers the pressure, the water is too hot for the lower boiling point and the extra heat converts some of it into steam."

"That's it. You've got it exactly."

Pete was a pretty happy lad in his new-found knowledge, but before I let him go I got a textbook out of my desk, found a good steam table and piloted him through some figuring. "Now, Pete, I think you have it and it's time we were getting home to our dinners, but before we go I want to show you something. Come here," and I took him to the front of a boiler. I blew out the water column so as to fill it with fresh hot water from the boiler. "Now if I shut off the top and bottom connections to that glass and they are tight, the steam in the top half will condense and lower the pressure in the glass and the water in the bottom half will boil as it gives off its surplus heat. Try it," I suggested. Pete fairly jumped at those valves, but it wouldn't do to print what he said when that water began to boil and the bubbles of steam started rising to the top.

"Say, Chief," said Pete as we started to leave, "a pop safety valve ought to blow longer when the glass is full."

"Yes, Pete, and a boiler under a load with sudden heavy fluctuations should keep the steadiest pressure when the glass is well filled."

"Good night, Pete."

"Good night, Chief."

Next morning when I came on duty I found a box of —well never mind what, they are all gone now anyway— on my desk labeled in lead pencil "Stored Energy."

Vancouver, B. C., Canada. R. MANLEY ORR.

[Considerable space is devoted by Professor Thurston in his book, "The Manual of Steam Boilers," to the subject of the energy stored in the hot water in a steam boiler under high pressure. While his discussion is particularly in relation to boiler explosions, the results are equally applicable to the subject dealt with in the foregoing letter. Without attempting to go into the subject fully, some quotations will be of interest.

Very little of the destructive effect of an explosion is due to the steam which is confined in the steam chamber at the moment of the explosion, for as soon as the steam escapes and thereby diminishes the compressive force upon the water, a new issue of steam takes place from the water by reducing its temperature; when this escapes and further diminishes the compressive force, another issue of steam of lower elastic force from the water takes place, again reducing its temperature; and so on until at length the temperature of the water is reduced to the atmospheric boiling point and pressure. Thus it is shown that it is the enormous quantity of steam so produced from the water during this continuous but exceedingly rapid operation that produces the destructive effect in steam-boiler explosions and the action of the steam which may happen to be present in the steam space at the instant of rupture is considered unimportant. The quantity of energy stored in the water is vastly in excess of that contained in the steam, notwithstanding that the amount of energy per unit of weight of fluid is enormously greater in the steam. Nevertheless the disproportion of weight of the two fluids in a boiler makes the total energy stored in the steam quite insignificant in comparison with that contained in the water. The energy stored in steam boilers is capable of very exact computation, and the results reached in some cases are given. A locomotive boiler rated at 650 hp. under 125 lb. pressure, containing 6,900 lb. of water and 31.5 lb. of steam, has stored within it 61,148,790 ft.-lb. of energy in the water and 2,135,802 ft.-lb. in the steam, making 71,284,592 ft.-lb. in all. A smaller boiler of the same type under the same pressure, rated at 425 hp. and containing 6,330 lb. water and 19.2 lb. steam, has stored in it 64,235,160 ft.-lb. of energy in the water and 1,302,431 ft.-lb. in the steam, or a total of 65,555,591 ft.-lb.

With increasing steam pressure and the unremitted growth of these magazines of stored energy, today the amount of available energy is often as much as 200,000,000 or even 300,000,000 ft.-lb. in a single unit, and the total energy, pound for pound, is only comparable with that of gunpowder.

While the foregoing figures are interesting, they also form a convincing argument in favor of keeping every part of a pressure vessel in "shipshape," or in the best possible working condition, to avoid the risk of the penalty too often paid for carelessness or inefficiency in the operation of steam boilers.—Editor.]

Brass Polishing Paste

In regard to the formula for making brass polish, on page 54 in the issue of July 11, I think that I have a better and cheaper polish, which is very simple and can be made in any desired quantity. I have used it for three years.

Mix Bon Ami powder and ammonia to the consistency of cream, rub it on, let it dry and polish with a soft cloth.

Cohoes, N. Y. W. E. PARLEY.

Two- vs. Four-Stroke-Cycle Marine Diesels

In the same issue with my last letter (June 6, 1916) on "Heat Maladies of Marine Diesels," there was a communication from H. Schreck containing the plea that those concerned with the Diesel engine in this country unite in their engineering endeavors instead of condemning. That statement was no doubt made with good intent, but does it express a rightful attitude? I believe if unity of endeavor is to be obtained at the cost of withholding condemnation from that which is to be condemned, then the unity is not desirable.

Prejudices have been created here against the Diesel-type engine primarily by failures of certain engines made in this country, but also by the lack of discussion of these failures. If reason be shown for the failures of certain designs of any special class of machinery, then it is possible to gain appreciation for the success of other designs of that same class of machinery, whereas if the failures are allowed to remain as undisturbed recollections in the minds of engineers, then one must expect that the whole class of machinery will be branded as unfit.

The plea of Mr. Schreck would be entirely wholesome if all Diesel-type engines were fitted for trustworthy service, but so long as there are manufacturers ready to throw the burden of experiment upon guileless purchasers, there cannot be this unity of endeavor.

In some respects Mr. Schreck's letter was likely to mislead. Who would not have concluded from his reference to the motor ship "Secundus" that all was well with that vessel? As a matter of fact, when the story is continued from the point where he left off, a wholly different impression will be created. From New York the vessel went to New Orleans, where she had certain repairs made which were not very important, although they were pressing, but when her engines were opened up in Hamburg after the round voyage had been com-

pleted, there were found to be four cracked cylinder heads and all the bearings were squashed down and out of shape, so as to necessitate realigning the crankshaft and pouring new white metal into the crankpin boxes and crosshead pin boxes as well as into the main-bearing shells.

Each engine had been stopped every day for an examination that had become wholly necessary. The lubricating-oil consumption had been 1,050 lb. per day on the outward passage and 950 lb. per day on the homeward passage. By weight this was equal to about 4 per cent. of the fuel burned and was equivalent to 25 or 30 per cent. of the latter in cost. Yet this grief came with engines of about 4,600 i.hp. that were operated at about 3,400 i.hp. outward and at about 2,600 i.hp. homeward.

The "Rolandseck," to which Mr. Schreck pointed as another instance of a successful two-stroke-cycle engined ship, has also been constantly beset with troubles, although her voyages are comparatively short. A report from her engineer was published in German technical periodicals and in *The Engineer* (London) and showed the same story of bearing troubles, cracked cylinder heads, seized fuel valves, faulty piston cooling, etc., which can be written down as inseparable evils of the two-stroke-cycle engine.

To the bearing problem in two-stroke-cycle engines Mr. Schreck particularly refers in another paragraph, expressing the opinion that designers should be able to overcome this difficulty. No matter what the future may have in store, the present has shown that no designer has yet solved the problem. In the very small sizes of two-stroke-cycle engines some measure of successful service has been obtained, because the proportions of the parts are not so unfavorable as in the larger engines, and also there are some stationary engines of large power operating on the two-stroke cycle without serious trouble, favored by the conditions of operation.

Cracked heads on four-stroke-cycle engines are mentioned by Mr. Schreck. They do occur occasionally, but are more likely to occur in medium-sized stationary engines than in the large marine engines.

I also take issue with Mr. Schreck on the importance of openings in cylinder heads. He states: "As more material can be provided where there are only two openings instead of four, which cover does one suppose will resist the heat best?" It is clear that the head with two openings is intended to refer to a two-stroke-cycle engine, whereas that with four openings was meant to refer to a four-stroke-cycle engine.

The best head in Diesel-engine design will have four openings or more, because the best four-stroke-cycle engines are built with such a design of head, and the head of such an engine can resist the heat much better than can one of a two-stroke-cycle Diesel engine. Mr. Schreck must know that the number of openings in a cylinder head is in any case a minor factor, far more important being the amount of heat to which the head is subjected and also the manner in which the openings are designed. One bad opening in a cylinder head is far more undesirable than six good openings.

Mr. Haynie in the same issue refers to a pair of 1,650-hp. two-stroke-cycle marine engines he saw in Sweden and states that it is certainly to be admitted that an engineer having the number of years' experience that Mr. Hesselmann had had in building both two- and four-stroke-cycle engines is capable of judging whether the two-stroke-cycle marine engine can be made a success. Mr. Haynie went astray. Mr. Hesselmann made a serious failure of his two-stroke-cycle engines for the motor tank ship "Sebastian." They had to be removed and were replaced by four-stroke-cycle engines. His 1,650-hp. engines were in course of building before the "Sebastian" fiasco happened, otherwise they might not have been built.

Further remarks of Mr. Haynie anent the fair division of business between the two-stroke-cycle and the four-stroke-cycle engines raise again the question to which the opening paragraphs of this letter refer. By all means let manufacturers try to develop the two-stroke-cycle engine, but also let it be made clear that there has hitherto been a continuous list of failures in the wake of that type of marine engine, and this should be sufficient warning to prevent the cost of experiments being heaped upon unsuspecting buyers in the future, as it largely has been in the past. R. W. CROWLY.

Jersey City, N. J.

Hammering Water Pipes

One morning the office manager called up the power house and complained that the pipes were making an awful racket. We found that the hot-water pipes were hammering severely at one point, at the end of a line. The master mechanic suggested that perhaps a valve was gone in one of the pumps, resulting in an irregular flow as the cause of the hammering, but the pumps were found to be in good shape when the valves and piston packings were examined and a couple of pistons repacked, yet the hammering continued as bad as ever. It was partly relieved by leaving a valve near the end of the line, where the worst of the hammering took place, partly open, allowing a small stream of water to escape all the time. This makeshift was the only remedy we could find for a couple of weeks. One day in passing one of the pumps I noticed water dribbling down the side of the air chambers of a 2,000-gal. flywheel pump, used to circulate the hot water.

The pet-cock was partly open, allowing the air to escape and the air chamber to fill with water. The oiler, who was a new man, said he had opened the cocks to keep the air out of the pumps. After closing the pet-cocks and explaining to the oiler the purpose of the air chamber, I opened a small valve in the suction line a little to again fill the air chambers with air. I went over to the office and noticed that there was no hammering at all in the pipes. I closed the bleeder valve that we had left open, but this made no difference. The cause of the trouble then became clear to me. The big pump was nearest to the office and was its main supply, although it was tied in with the other hot-water pumps on the system, and the lack of air in the air chamber had been the cause. To determine if this was correct, I went back to the pump and let the air blow out, then went back to the office and found the pipes were hammering as bad as ever. I hurried back and filled the air chamber again, and pretty soon the racket stopped altogether. The pump, being of the flywheel type and running about 60 r.p.m., had tended to make it slam all the more. E. W. MILLER.

Minneapolis, Minn.

Inquiries of General Interest

Shortage of Ammonia—What are the indications of a shortage of ammonia in a compression plant? C. B.

The level of liquid in the gage-glass of the receiver will be low, the valves of the compressor will become noisy, the expansion valve will also be noisy due to gas going through with the liquid, and the head pressure will fluctuate.

Ground on Primary of Transformer—In a service transformer having a 2,300-volt primary and 110-volt secondary, with one side of the primary grounded and the secondary insulated, will the primary voltage have any effect upon the secondary circuit in case a short-circuit occurs between the primary and secondary windings? T. A. C.

There will be no effect as long as the secondary is insulated, but as soon as a ground occurs on the secondary, the primary fuses or circuit-breaker will open. This condition would be dangerous to have on any system, for if an attendant came in contact with the low-voltage system and the ground at the same time, it would be practically the same as getting across the high-voltage line. Many serious accidents have occurred from this cause. When the transformer is installed, not only should the case of the transformer be grounded, but also the middle point of the low-voltage winding. If the middle of the secondary winding is not brought to the outside of the transformer, one of the low-voltage leads should be grounded.

Proximate Analysis of Coal—How is the proximate analysis of a fuel made, and to what extent are the results of value in determining the combustible value of coal? W. F.

The proximate analysis gives the proportions of fixed carbon, volatile combustible, moisture and ash, and affords information as to the general characteristics of a fuel and its relative heat value. The proximate analysis is made by subjecting a sample to a temperature of from 250 deg. F. to 300 deg. F. to expel the moisture; then to a red heat, which expels the volatile matter; then to a white heat, which causes the carbon to pass off as dioxide, leaving the ash as a residue. By weighing the residue at the end of each operation, the various percentages can be computed. The heat value of the fuel cannot be directly computed from the proximate analysis, as the volatile content varies in composition and heat value. Therefore the proximate analysis does not furnish information for computing the heat of combustion with the same degree of accuracy as an ultimate analysis, but estimates may be based on the proximate analysis that are sufficiently correct for most practical purposes in comparison of different lots of the same kind of coal.

Allowable Working Pressure for Dome Head—What working pressure would be allowable for the head of a steam dome 36 in. diameter, bumped to a radius of 40 in., made of $\frac{3}{8}$-in. steel of 55,000 lb. tensile strength, without manhole or other opening, and double-riveted to the dome shell? A. A. R.

The thickness required in an unstayed dished head with the pressure on the concave side, when a segment of a sphere, may be calculated from the formula,

$$t = \frac{5.5 \times P \times L}{2 \times TS} + \frac{1}{8} \qquad (1)$$

where

t = Thickness of plate, in.;
P = Maximum allowable working pressure, lb. per sq.in.;
TS = Tensile strength of the material, lb. per sq.in.;
L = Radius to which the head is dished, in.

By transposing and reducing, the formula becomes

$$P = \frac{(2t - \frac{1}{4}) \times TS}{5.5 \times L} \qquad (2)$$

and by substituting in (2) the values stated,

$$P, \text{ the maximum allowable working pressure} = \frac{[(2 \times \frac{3}{8}) - \frac{1}{4}] \times 55{,}000}{5.5 \times 40} \text{ or } 125 \text{ lb. per sq.in.}$$

Weight Required for Lever Safety Valve—What weight would be necessary for a lever safety valve to cause it to blow off at 100 lb. if the diameter of the valve were 4 in., weight of valve and stem 5 lb., weight of lever 12 lb., distance from the fulcrum to the center of the valve 4 in., distance from the fulcrum to the center of gravity of the lever 15½ in. and distance from the fulcrum to the center of the weight 36 in.? B. E.

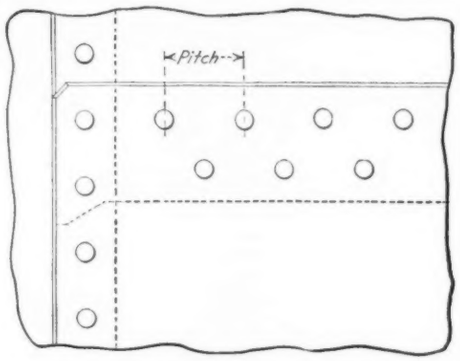
The area of the valve would be $4 \times 4 \times 0.7854 = 12.566$ sq.in., and the total pressure of steam on the valve would be $12.566 \times 100 = 1{,}256.6$ lb. This would be balanced in part by the weight of the valve (5 lb.) and the force exerted by the weight of the lever, which for a lever weighing 12 lb., a fulcrum distance of 4 in. and distance from the fulcrum to the center of gravity of the lever of 15½ in. would be $(12 \times 15.5) \div 4 = 46.5$, so that the net force required of the weight would be $1{,}256.6 - (5 + 46.5) = 1{,}205.1$ lb. As the distance from the fulcrum to the valve is 4 in. and the center of the weight would be placed 36 in. from the fulcrum, each pound of the weight would press upon the valve with a force of $36 \div 4 = 9$ lb., and as a force of 1,205.1 lb. would be required, the safety-valve weight would have to be $1{,}205.1 \div 9 = 133.9$ lb.

Efficiency of Double-Riveted Lap Joint—What would be the efficiency of a double-riveted lap joint made of $\frac{5}{16}$-in. plate of 55,000 tensile strength, with $\frac{3}{4}$-in. rivets pitched 2⅞ in.? C. H.

With the thickness of plate of $\frac{5}{16} = 0.3125$ in., pitch of rivets $2\frac{7}{8} = 2.875$ in., and tensile strength of plate = 55,000 lb. per sq.in of cross-sectional area, the strength of solid plate per unit of pitch (indicated in the sketch) would be,

(A) $2.875 \times 0.3125 \times 55{,}000 = 49{,}414$ lb.

The strength of the joint would depend on the strength of plate between the rivets; or the shearing strength of the



DOUBLE-RIVETED LAP JOINT

rivets; or the resistance which the plate offers to the crushing action of the rivets.

Having the rivet $\frac{3}{4}$ in. diameter, the cross-sectional area of plate between rivet holes would be $(2.875 - 0.75) \times 0.3125$, and, per unit of pitch, the strength of plate between rivet holes would be,

(B) $(2.875 - 0.75) \times 0.3125 \times 55{,}000 = 36{,}523$ lb.

The strength of rivets per unit of pitch would depend on the strength of two rivets in single shear. Allowing a shearing strength of 44,000 lb. per sq.in. of cross-sectional area, as the cross-sectional area of each ($\frac{3}{4}$ in. diameter) rivet would be 0.4418 sq.in., the shearing strength of rivets per unit of pitch would be,

(C) $2 \times 44{,}000 \times 0.4418 = 38{,}878$ lb.

The crushing action of the rivets per unit of pitch would come from two rivets, and allowing a crushing strength of plate of 95,000 lb. per sq.in., the crushing strength of plate in front of two (2) rivets would be,

(D) $2 \times 0.75 \times 0.3125 \times 95{,}000 = 44{,}531$ lb.

As the strength of the joint per unit of length would depend on (B), the strength of plate between rivet holes, the efficiency of the joint as compared with the solid plate would be,

$$\frac{(B)}{(A)} = \frac{36{,}523}{49{,}414}, \text{ or } 73.9 \text{ per cent.}$$

[Correspondents sending us inquiries should sign their communications with full names and post office addresses. This is necessary to guarantee the good faith of the communications and for the inquiries to receive attention.—Editor.]

Status of Pending Water-Power Legislation

Impressed by the country's need for cheap power for use in new industries and extensions of existing industries in which power constitutes a large factor in cost of production, the conference committee of the Senate and House is endeavoring to reach a compromise agreement on a general dam bill that can be approved by both houses and become law before the end of the present session of Congress.

Both branches of Congress have passed bills for the granting of permits for power dams and plants in navigable streams. The House measure contains a number of provisions placing more drastic restrictions upon development enterprises than are provided in the Senate bill. The difference in the two measures which the conferees expect to be the most troublesome, however, is that relating to the manner in which power permits shall be granted.

The Senate bill proposes that the Secretary of War shall have authority to grant such permits, under uniform and general terms, conditions and regulations prescribed in the measure. The House bill provides terms and conditions under which dams may be built, to be administered and enforced by the Secretary of War, with the condition that before any plans shall be approved or agreements entered into for the construction of such dams, Congress shall in each case pass a special enabling act granting its consent to the location and construction. This is the procedure provided under the present general dam act. In operation, those who favor giving authority directly to the Secretary of War say, the present system has not been found to work well. No grants of any kind have been made by Congress since 1912, while in the six years before that date in which the law was in operation, about thirty enabling acts were passed by Congress, but only four comparatively small power plants were built under them, it being impossible in most cases to finance the projects under the terms of the present law.

The Senate bill provision for permits being granted by the Secretary of War is in harmony with the present method of authorizing power constructions in national forests, national parks and other parts of the public domain, for which permits are issued by the Secretaries of Agriculture and the Interior.

COMPROMISE LEGISLATION PROBABLE

There are several other differences of policy in the Senate and House bills on which the conference committee will try to find a middle ground that will make compromise legislation possible. Some of these differences hinge mainly on the wording of particular sections or clauses, as in the case of the so-called "recapture" section, providing the procedure and terms on which the Government may take over power plants at the end of fifty years. Other differences, which are likely to be more difficult of compromise, have to do with the question of Government charges and restrictions upon the business of grantees.

The Senate bill requires the grantee to construct any locks and other aids to navigation which may be required by the Secretary of War at the time the dam is built and to operate such locks, etc., without expense to the Government. The House bill requires that the grantee shall construct such locks as the Secretary of War or Congress may "at any time" deem to be necessary. This requirement, if retained in the law, it is pointed out by its opponents, would make impossible the financing of a power dam and works, since the grantee would never know when the Government might demand that he invest additional large sums of money, the interest on which would add to the cost of power and might make it impossible to carry out without a loss contracts previously entered into.

Both bills require the grantee to pay a charge for whatever increased water power may be created by improvements constructed by the Government for storage of headwaters of streams, and both bills provide that the grantee shall maintain lights, signals, fishways, etc., and shall pay a tax or rental for use of surplus waters where a power plant is built at a Government dam. In addition to these charges the House bill gives the Secretary of War authority to require the grantee to pay a tax on the grant itself, the amount of which is left to the discretion of the secretary and which may be readjusted by him at the end of twenty years and of each ten years thereafter. Water-power men say that such a requirement would go far to discourage development enterprises, since every additional cent of charges placed on water power adds to its cost and must be added to the rates consumers have to pay, thus minimizing the economy of water power over steam. To regulate rates, as provided for in both bills, and also to tax the right to generate power, they say, is inevitably creating a tax that must be transferred to the consumer, while the proposal to readjust taxes periodically creates uncertainty concerning the future costs and profits of the company and would discourage investors from putting money into water-power enterprises.

Both bills provide that water-power grants shall be irrevocable for fifty years, except for cause, and that at any time after the expiration of fifty years the Government may take over the property, or authorize a new grantee to do so, at a valuation to be mutually agreed upon or fixed by the courts, with no allowance for any rights granted by the Government. The differences in this valuation provision are largely a matter of phrasing and definition. The Senate bill, however, proposes that after fifty years the original grantee shall continue in possession, under the original terms and conditions, until the Government decides to take over the property or has a chance to make a better bargain with a new grantee. The House bill terminates the grant absolutely at the end of fifty years and provides that the Secretary of War then make a new grant upon such terms and conditions as existing laws may then require, or may do nothing.

In view of the declaration favoring use of natural resources, contained in the St. Louis platform, it is believed that the President and Democratic leaders will make every effort to secure the passage at this session of a water-power bill that will bring about development, and it is expected that an agreement on such a measure will be reached by the conference committee on the navigable streams bill before adjournment.

California Engine Safety Orders

The following orders have been issued by the Industrial Accident Commission of California and went into effect Aug. 1:

Order 100. (a) All stationary steam engines, gas engines, air compressors, electric generators and pumps must have all gears, belts, pulleys, clutches, shafts, keys and keyseats, collars, setscrews and sprockets guarded according to the orders as set forth in General Safety Orders issued by this commission. (Portable engines and pumps, such as logging engines, portable hoisting engines and engines and pumps used in construction work, are not classed as stationary engines.)

Order 101.[1] (a) Each engine must be equipped with an effective governor which will at all times automatically control the speed of the engine under varied loads, except where the load itself acts as an effective governor. All belt-, rope- or chain-driven governors must be equipped with a safety device which will stop the engine in case the belt, rope or chain should break.

(b) Projecting rotating parts of all engine governors shall be provided with substantial guards.

Order 102. (a) Valve gears must be so arranged, or other provisions made, that in the event of the load being removed, the engine will stop if the governor fails to act. (A broken governor-belt stop will be considered sufficient for slide- or four-valve engines.)

Order 103. (a) All parts of engine room not accessible from the floor and requiring attention must be provided with adequate runways, rough-tread platforms or rough-tread stairways.

(b) Where stairways are not practicable, fixed ladders must be installed, the rounds of which shall be not less than 6 in. in clear from wall or column. (The use of metal ladders is strongly advocated.) (See editorial, page 247.)

(c) The protection of platforms and runways shall be in accordance with Order 13 of the General Safety Orders.

Order 104. (a) No repairs must be made on any vertical or horizontal engine, compressor or pump until the engine has been securely blocked or other provisions made to prevent revolving or reciprocating parts from turning over accidentally.

Order 105. (a) Wherever floors, stairways or platforms in engine rooms become slippery, they shall be equipped with rough surface plates or treads.

Order 106. (a) In engine rooms all floor openings must be protected by a substantial covering or be guarded in accordance with Order 17 of the General Safety Orders. (The use of rough-tread metal plates is advocated for floor-opening covers.)

Order 107. Flywheels on stationary steam engines, gas engines, air compressors, electric generators and pumps must be guarded as follows:

(a) If guard is at least 15 in. and not more than 18 in. in the clear from each side and face of the wheel, a fence may be used at least 3½ ft. high, consisting of two rails, the bottom rail of which must be at a point 18 in. from the floor, and no rails shall be spaced a greater distance than 24 in. between centers.

[1] It is recommended that each engine shall be equipped with an automatic stop, independent and a complete unit in itself.

(b) If guard is less than 15 in. in the clear from each side and each face of the wheel, a substantially supported wire-mesh or close-slat guard with openings not greater than 2 in. in any direction must be provided at least to the top of the wheel, if the wheel does not extend over 5 ft. in height, and in no case shall this guard be less than 3½ ft. high. If the top of the wheel is more than 5 ft. in height, the guard must be at least 5 ft. high.

(c) All flywheel pits must be surrounded with toeboards not less than 6 in. high.

(d) In case the above method of protection is not desired, flywheels must be housed completely with substantially supported wire mesh, close slats or solid material.

(e) Where it is necessary to move flywheels for starting, guards may be removed temporarily, but must be replaced immediately after such operation is complete.

(f) All rope-drive wheels and pulleys from main engine to jackshaft must be covered with a substantial housing to prevent lashing of broken rope.

(g) Every engine transmitting power by means of a rope drive must be equipped with a telltale which will give ample warning of the stranding of the ropes.

Order 108. (a) All cranks and connecting-rods of center-crank engines and of high-speed side-crank engines shall be equipped with a complete and substantial metal housing, which may be provided with doors where necessary.

(b) All slow-speed side-crank engines shall be provided with a guard at least 15 in. in the clear from the crankpin, and at least 3½ ft. high.

Order 109. (a) Two exits shall be provided for each engine room, said exits to readily open from the inside without the use of a key.

(b) In engine rooms below ground level, at least one exit must be isolated from the engine room by a fireproof inclosure the entrance of which must be provided with a door at floor level.

Note—The intent is to protect operators against scalding or suffocating vapors or gases.

(c) All engine rooms shall be properly ventilated to prevent the accumulation of noxious vapors, odors and fumes.

Order 110. (a) All trenches, tunnels and pits in engine rooms and leading therefrom shall have proper drainage and ventilation and be kept in a sanitary condition.

(b) All sumps and sewage tanks in engine rooms shall be tightly covered and ventilated to the outside atmosphere, using a vent not less than 4 in. diameter and so located as not to endanger the safety of anyone in the vicinity.

Order 111. (a) All traps, drains and trap discharges from engines shall be piped to a line, blowoff tank or other safe place of discharge.

(b) The discharge of all engine exhaust pipes must be located so as not to endanger the safety of anyone in the vicinity.

Recommendations—It is recommended that every ammonia refrigerating plant maintain in good order a helmet or other similar device approved by this commission for use in case it is necessary to enter the room in the presence of excessive ammonia fumes, and that this device be kept in close proximity to the entrance and on the outside of the compressor room.

Water-Power Resources of the Winnipeg River

The report by J. T. Johnston, chief hydraulic engineer of the Dominion Water Power Branch of the Department of the Interior, of Ottawa, Canada (Water Resources Paper No. 3), shows that there will eventually be as much power available from the Winnipeg River as is now being developed at Niagara, if not more. What this represents can be realized only by one who knows the tremendous import of electro-chemical and electro-metallurgical industries of worldwide fame now established at Niagara, and by one who has the proper conception of the significance of the use to which Niagara power is now put. The potential powers of the Winnipeg River are not generally appreciated. There are at present five large power and industrial plants in operation, namely:

The Winnipeg Electric Railway Co.'s 34,400-hp. plant consisting of four units of 2,600 hp. rating and five of 4,800 hp. each.

City of Winnipeg's 47,600-hp. municipal plant containing five units of 5,200 hp. and three of 7,200 hp. each.

Municipal plant of Kenora, consisting of four turbines of 900 hp. and two smaller wheels for the exciter units—over 3,600 hp. in all.

Lake of the Woods Milling Co.'s plant of approximately 5,800 hp. at Keewatin has two 2,400-hp. turbines, one vertical 95-hp. and two horizontal turbines of 360 hp. each and one of 180 hp. rated. At Mill "C" 1,400 hp. can be put out.

The Ontario & Minnesota Power Co., at Fort Francis, Ottawa, and International Falls, Minn., was laid out for a 36,000 turbine horsepower total, but this calls for a much greater flow than the river affords. The power from this plant is utilized by the local paper mills.

The complete report, covering the entire water-power situation, developed and undeveloped, consisting of some 500 pages in Vol. 1 and in Vol. 2 about 100 pages of topographic sheets of land surveys in considerable detail, may be obtained by applying to the superintendent of the Dominion Water Power Branch, Department of the Interior, Ottawa, Canada.

N.A.S.E. Delegates' Photographs

At the national conventions of the National Association of Stationary Engineers, "Power" gives away as a souvenir the "Power Pup," a comic daily paper. The paper is free and contains no advertising; it's a fun maker pure and simple.

On the back page of each day's "Pup" appear individual pictures of the delegates attending the convention. Many are still to be heard from, and we take this occasion to advise each delegate that the post-card photograph of himself (bust only) should be received at this office not later than Aug. 28 to insure getting it in the "Pup." Delegates are given the electrotype when applying at our booth at the convention.

Fatal Engine Wreck

At about 9 o'clock in the evening, Aug. 3, the cylinder head of a 14x24-in. Corliss engine at the plant of A. Colburn & Co., 110 North Second St., Philadelphia, Penn., was knocked out and broken into fragments, killing the engineer, Silas Batton. The victim, who happened to be passing just as the accident occurred, had both legs broken and was terribly scalded. Considerable time elapsed before anyone could be found to shut off the steam.

The cause of the disaster is supposed to have been a flood of water from the boilers in the cylinder, as the piston was pulled off from the rod and the cylinder-head studs were broken off flush with the end of the cylinder. There was no separator in the steam line (nor an emergency quick-closing cutoff valve, evidently). The property loss was considerable.

Minneapolis Armory Not To Be Used for Exhibit

James F. Gould, secretary of the local committee which is arranging for the coming national convention and exhibit of the National Association of Stationary Engineers at Minneapolis, Sept. 11 to 16, advises that the armory, which was to house the exhibit, was last week closed to the public by the building inspector. The exhibit and convention will be held in the Pence automobile warehouse, Eighth Avenue north and Washington Avenue.

The exhibit will be on the first floor, and convention business will be conducted on the second floor. The Pence Building has more floor space and is nearer the business section of the city than the armory.

The trip to South St. Paul, which was to be made by the conventioners on Thursday, Sept. 14, has been abandoned on account of lack of adequate transportation facilities.

PERSONALS

George E. Emmons, for twenty years plant manager of the General Electric Co., at Schenectady, N. Y., was recently elected vice-president of the company.

A. D. Quackenbush, formerly of the engineering department of H. M. Byllesby & Co., of Chicago, and more recently assistant superintendent of the Mobile Electric Co., Mobile, Ala., has been appointed general superintendent of the latter company.

William A. Evans recently took charge of the New York office, 149 Broadway, of the Whitlock Coil Pipe Co., Hartford, Conn.

E. Burt Fenton, manager of the publicity department of W. S. Barstow & Co., Inc., has moved his headquarters from Sandusky, Ohio, to the company's general offices at 50 Pine St., New York City.

M. S. Wilson, for 25 years head of the general testing department of the Fort Wayne Electric Works of the General Electric Co., at Fort Wayne, Ind., has been appointed motor and generator inspector.

B. O. Austin, who was formerly connected with the engineering department of the Westinghouse Electric and Manufacturing Co. and more recently with Gilbert C. White, consulting engineer, Durham, N. C., has opened an electrical, civil and hydraulic engineering office at Charlotte, N. C.

Recent Court Decisions

Digested by A. J. H. STREET

Duty Concerning Electric Wires—Companies maintaining highly charged electric wires in places where it may be reasonably expected that persons will be exposed to danger therefrom must exercise a very high degree of care to keep them in safe condition. But the law does not exact as high degree of care in the maintenance of dangerous machinery or appliances in an unfrequented place as is required along or across public highways. The measure of care is what is usual and customary under similar circumstances. The unexplained breaking of an insulator, causing a highly charged electric wire to sag to within six or seven feet of the ground, at a place remote from any public highway, and its remaining in that condition for a day do not prove negligence on the part of the electric company, in the face of direct testimony that reasonable care was exercised to prevent such accidents and to discover and remedy such situations. (West Virginia Supreme Court of Appeals, Owen vs. Appalachian Power Co., 89 "Southeastern Reporter," 262.)

Liability Concerning Uninsulated Wire—An owner of high-voltage wires who negligently permits a wire to remain uninsulated for several years after the original insulation has rotted off is bound to foresee the possibility of a boy throwing a small wire over the line, and may be held liable for injuries sustained by a woman who rushes to the aid of a boy electrocuted through contact with the wire. (Kansas Supreme Court, Snyder vs. Leavenworth Light, Heat and Power Co., 157 "Pacific Reporter," 442.)

Rights of Water-Power Companies—A water-power company must pay damages to a county for overflow of its public highways, unless the legislature in granting a franchise to the company to maintain a dam has provided otherwise. (South Carolina Supreme Court, Edgefield County vs. Georgia-Carolina Power Co., 88 "Southeastern Reporter," 801.)

Duty to Safeguard Shafting—Under the Nebraska statute that requires shafting to be safeguarded for the protection of employees required to work near it, the supreme court of the state has affirmed judgment for $20,000 in favor of an employee in a pumping plant who was seriously injured when caught by an unguarded coupling on overhead shafting while obeying instructions from his superior to whitewash the walls of the plant. (McCarthy vs. Village of Ravenna, 157 "Northwestern Reporter," 629.)

Warranting Efficiency of Machinery—In a suit by the purchaser of draft-inducing apparatus to recover damages from the seller for claimed breach of warranty as to the efficiency of the apparatus, the seller is not entitled to rely upon any inaccurate information given by the buyer as to existing draft if the misinformation did not influence the installation of a plant of the required power. A consulting engineer was qualified to testify to the value of the apparatus in its actual condition and as warranted, on proof that he was an experienced engineer and had bought 16 similar plants. (United States Circuit Court of Appeals, Sixth Circuit, Champion Fiber Co. vs. B. F. Sturtevant Co., 232 "Federal Reporter," 1.)

Representations Concerning Appliances—In a suit to recover the price of superheaters installed in a power plant, defended by defendant under claim that misrepresentations were made to defendant concerning the equipment, it is held that in order to constitute fraudulent misrepresentations it is not essential that the person making them either knew that they were untrue or made them recklessly and consciously ignorant whether they were true or untrue; it is sufficient that the representations be false in fact and that the party making them be a party to the contract and profit by the buyer's loss. (Michigan Supreme Court, Power Specialty Co. vs. Michigan Power Co., 157 "Northwestern Reporter," 408.)

BUSINESS ITEMS

Sun Prairie, Wis., is in the market for a 65-hp. noncondensing engine for direct connection to a 40-kw. 250-volt d.c. generator. F. I. Conner is village clerk.

The American Steam Gage and Valve Manufacturing Co. has moved its Chicago office and storeroom to 25-27 South Jefferson St., where the manager, Charles C. Kilander, will have a larger and better stock to serve customers promptly and satisfactorily.

The Nordberg Manufacturing Co., of Milwaukee, states that it now has on order or building a large number of heavy-duty hoists and also is building a number of Nordberg-Carels Diesel engines for various mining companies in the West and Southwest.

Boiler Code Valves—To the list of manufacturers prepared to supply valves to meet the requirements of the A. S. M. E. Boiler Code, as published in the May 2 issue, should be added The Edward Valve and Manufacturing Co., Chicago.

The Bruce-Macbeth Engine Co. reports orders recently received aggregating 785 hp., many of which are second and third repeat orders. This shows that, contrary to prevailing opinion, the gas engine is not at a standstill, in the natural-gas field anyway.

The Moore Steam Turbine Corporation, of Wellsville, N. Y., has been organized with an authorized capitalization of $160,000, to build an improved type of single and multi-stage steam turbine in sizes from 5 to 1,000 hp. The officers of the corporation, who are all men of long experience in the steam-turbine business, are as follows: J. L. Moore, president, formerly chief engineer of the Kerr Turbine Co.; J. B. Laird, vice-president, formerly factory manager of the Kerr Turbine Co.; E. D. Spicer, secretary and treasurer, formerly chief draftsman and acting chief engineer of the Kerr Turbine Co. Plans have been drawn and the contract will be let at once for a modern fireproof reinforced-concrete factory building, power house and testing plant.

TRADE CATALOGS

Nordberg Poppet-Valve Engines—Bulletin No. 28, recently published by the Nordberg Manufacturing Co., covers a line of poppet-valve engines which includes three types—full poppet, poppet-uniflow and poppet-Corliss. It is pointed out that for ordinary noncondensing service, the high-speed full poppet-valve engine shows the highest efficiency. For condensing service two types are offered, the poppet-uniflow and poppet-Corliss. The former is particularly adapted to widely fluctuating loads on account of its flat steam-consumption characteristic. Recently two of the largest uniflow engines so far constructed in this country have been shipped to the Youngstown Sheet and Tube Co. for steel-mill drive. The poppet-Corliss engine is adapted to constant load or constant m.e.p. work. A compound engine of this type, fitted to an ammonia compressor, showed by test a steam consumption of less than 10 lb. per indicated horsepower-hour. Following the discussion of the different types of engines and their construction, a number of pages are devoted to test results. Then comes a short discussion of the application of poppet-valve engines to compressors, pumps and ice machines. The remaining part of the bulletin is devoted to the illustration and description of the frame, bearings, rods, cranks and receiver.

"J-M High-Temperature Cements." H. W. Johns-Manville Co., New York City. Bulletin. Illustrated; 11 pp.; 3½x6 in. This bulletin tells of the advantages of using high-temperature cement, a heat-resisting compound that vitrifies at 3,000 deg. F. It is applied to the brick the same as mortar. The printed matter will be of interest to engineers who have had or are having trouble with their furnace linings after using cement or fire-clay as a binder.

Boiler Settings and Dimensions. Sanford Riley Stoker Co., Worcester, Mass. Booklet. Pages 32; 6¼x9 in. The purpose of this data book is to explain the adaptability of the Riley stoker. Numerous illustrations show some of the combinations that are possible in regard to air connections, ash removal, stoker drives and furnace arrangements. Dimensions are given that will assist in determining the space required for the installation of the stoker.

Smoke Preventer. Climax Smoke Preventer Co., 261 Franklin St., Boston, Mass. Catalog. Pp. 20, 6x9 in. Illustrated. This contains table of boiler test of boilers equipped with preventer and without preventer.

Insuluminum—Bulletin 118, just issued by the Diamond Power Specialty Co., Detroit, contains a number of photomicrographs and describes the properties of this new product of the General Electric laboratories, which will withstand a continuous temperature of 1,800 deg. F., as compared to 1,200 deg. F. for iron and steel. Photographs and records of a series of comparative tests at high temperatures of insuluminum and iron pipes, made by a number of well-known companies, are included. The bulletin deals primarily with the interesting new metal, but the application of the material to the Diamond mechanical soot blower system is described briefly on the concluding page.